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Analysis and Test Results for an Improved Constant Speed Passive Cyclic Pitch Wind Turbine

Final Subcontract Report

K. H. Hohenemser
Department of Mechanical Engineering
Washington University
St. Louis, Missouri

Prepared under Subcontract No. XE-2-02054-01

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PREFACE

This report was prepared by Dr. K.H. Hohenemser of the Department of Mechanical Engineering, Washington University. The work described here was performed under subcontract XE-2-02054-01 for the Solar Energy Research Institute (SERI) and was sponsored by the Department of Energy. The subcontract began on March 15, 1982, and with subsequent amendments ended December 31, 1985. During this period, 11 Progress Reports were submitted to SERI.

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SUMMARY

The research was to contribute to answering the question whether conventional blade pitch control of wind turbines with its high initial and maintenance cost can be replaced by rotor yaw control. The necessary high yaw rates are only possible with hinged or teetered rotors whereby the passive cyclic pitch (PCP) teetered rotor was found to be particularly suitable. In a PCP rotor the teeter axis forms a small angle with the blade axis. The research with an improved constant speed PCP wind turbine driving an induction generator made use of the test equipment at Washington University's Tyson Research Center available from a preceding study documented in the Final Report to SERI of December, 1981 (SERI/TR-1052-10). The atmospheric test equipment consisted of a horizontal wind turbine with vane controlled upwind two-bladed PCP rotor of 25 ft (7.6 m) diameter. The machine was mounted on a 60 ft (18 m) high Unarco Rhon SSV steel tower which was made tiltable for better access to the nacelle. The automatic rotor yaw control system was of a purely mechanical passive type. The rotor thrust which was laterally off-set from the yaw axis, in combination with a yawing component of the rotor torque due to uptilt of the rotor axis overcame at rated power the unfurling spring and furled the rotor with respect to the tail vane boom. The boom was attached to the nacelle by a pin located downwind of the yaw axis. Furling involved both nacelle rotation about the yaw axis and boom rotation about the pin. Due to the boom gravity moment, the boom pin Rulon bearings caused a large friction moment which resulted in imprecise delayed furling and unfurling.

In December, 1981 the boom pin was eliminated. The boom was instead supported by the yaw post and was moving together with the post. The nacelle, previously bolted to the yaw post, was now 90 degree rotatable about the yaw post with the help of two ball bearings. Thus tail vane boom yawing and nacelle furling were independent of each other and both functions involved little friction. This design principle was retained throughout the report period and it corrected previously observed imprecise delayed furling and unfurling.

A number of further modifications were found to be desirable. The blade airfoil was extended toward the rotor center with high twist which greatly improved start-up. Another improvement in the same direction was achieved when the shaft mounted gearbox with belt driven generator was replaced by an integral gear-generator unit used in the "Proof-of-Concept" version of the PCP turbine which reduced the break-out torque to one fifth of the previous value. Some erratic yawing at low wind speed power-on operation was corrected by an increased tail vane area and a lengthened tail boom which also corrected two dynamic conditions. The mechanical damper characteristics were found to be decisive for a proper functioning of the furl control. Many damper configurations were tested. As predicted by the analysis, an elastic restraint of the teeter motion reduced the negative rotor yaw damping and allowed a lower mechanical damper force.

As predicted by the trim analysis, a proper unfurl spring setting produced a power-off rotor speed that peaked at 1.3 times power-on rotor speed at 20 mph (9m/s) wind velocity. The autorotational rotor speed was lower at all other wind velocities. The PCP turbine was tested up to 60 mph (27 m/s) gusts. Power-off rotor speed was kept within narrow limits even in severe turbulence. From the test results one can expect smooth low load storm survival. Low wind speed generator cut-in and high wind speed generator cut-out were achieved by a digital controller in response to rotor speed without using wind speed information. Operation in deep stall was tested by

increasing the drive train transmission ratio. It was found that torque peaked at twice the value predicted by the conventional prop code.

Power-on operation in severe turbulence, though much improved by parameter optimization, was still less than desirable because of furl and power fluctuations with the natural tail vane period of several seconds. This period should be lengthened by aerodynamic or mechanical means. Alternatively, a slower reacting wind following system (for example a yaw control side rotor) should be used instead of the tail vane. The parametric testing was done on a trial and error basis and was hampered by the lack of an analytical yaw control dynamics model. Such a model should be developed and verified by tests with the PCP turbine after slowing down the wind-following system. Once a verified model was available, a generic study of torque and speed control by yawing of teeter rotors should be performed covering a spectrum of rotor parameters and control systems including active controls. The conclusion was that the PCP rotor continued to function very well up to high yaw rates and high wind velocities, but that the combination of the passive furl control system with a fast reacting wind-following tail vane caused some problems that could not be corrected as yet.

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NOMENCLATURE LIST

A	Rotor Disk Area
B	Number of Blades in Rotor
D	Rotor Diameter
H	Horizontal In-Plane Rotor Force
I	Blade Moment of Inertia
K	Teeter Spring Stiffness
N	Number of Samples Per Bin
P	Teeter Frequency with Teeter Springs over that Without Springs (in vacuum)
Q	Rotor Torque
R	Rotor Radius
S	Rotor Overhang (Distance to Yaw Axis)
T	Rotor Thrust
T_{EQ}	Equivalent Thrust (Eq. 3-1)
V	Wind Speed
$V/\Omega R$	Speed Ratio
a	Lateral Off-Set of Rotor Axis from Yaw Axis
$c_{0.7}$	Blade Chord at 0.7 R
c_H	Side Force Coefficient $(H/\rho A(\Omega R)^2)$
C_Q	Torque Coefficient $(Q/\rho A(\Omega R^2)R$
C_T	Thrust Coefficient $(T/\rho A(\Omega R)^2$
C_{TEQ}	Equivalent Thrust Coefficient
$dc_L/d\alpha$	Airfoil Lift Slope
β	Teeter Amplitude (Out-of-Plane)
γ	Blade Lock Number $(\rho\ dc_L/d\alpha\ c_{0.7}\ R^4/I)$

δ_3	Angle Between Blade Axis and Teeter Hinge
η	Rotor Axis Uptilt Angle
θ	Blade Pitch Angle
ρ	Air Density
σ	Rotor Solidity Ratio ($B\ c_{0.7}/\pi R$)
χ	Rotor Yaw Angle
Ω	Rotor Angular Speed

SECTION 1.0

INTRODUCTION

The horizontal axis wind turbine type under study incorporated the novel combination of two features; rotor passive cyclic pitch (PCP) variation and rapid rotor yaw angle variation to accomplish rotor speed and torque control by yawing. The wind turbine had a teetered rotor with 25 ft (7.6 m) diameter. It had a delta three angle of the teeter hinge (δ_3 = 67 deg) whereby the oscillation amplitude about the hinge axis was almost equal to the cyclic pitch amplitude about the blade axis. This type of rotor made rapid yaw rates possible without transferring gyroscopic or aerodynamic moments to the rotor hub and without causing excessive angular excursions of the blade tip path plane from the perpendicular to the rotor axis. Thus the hub could be easily yawed without resistance whereby the rotor plane followed the yawing motion of the hub almost instantaneously due to the aerodynamic servo action involved in the process. The experimental PCP turbine, installed April 1980 at Washington University's Tyson Research Center, was first operated with manual yaw controls, see Ref. 1. Rotor yaw rates up to 15 deg/s were encountered leading to cyclic pitch amplitudes up to $\pm$ 5 deg. The turbine was tested power-on at yaw angles of 0,15,30 and 45 degrees, power-off up to 80 degrees. Rotor loads and vibrations remained small. The alternator was tuned in such a way that torque increased with the square of the rpm so that the rotor would operate at constant wind speed with optimal tip speed ratio.

Two automatic rotor yaw control systems, one active and one passive system, were developed and tested in 1981, Ref. 2. Both systems provided an effective rotor speed and torque limitation. The passive rotor yaw control was simpler and it was expected to be more reliable. Except for some preliminary testing of the active system, all research up to the present was done in the configuration having the passive rotor yaw control. In September 1981, the variable speed alternator was replaced by a constant speed induction generator. Constant speed power-on operation was retained throughout the subsequent tests. It was found that with the turbulence level usually encountered at Tyson, variable rotor speed did not increase the captured rotor energy vs. that for constant rotor speed. However, for comparable wind speeds and turbulence levels, the dynamic rotor loads and the torque variations were less for variable rotor speed.

During the 1980/81 test period the tail vane boom was supported by the nacelle with the help of two Rulon bearings. This design caused excessive friction in the passive rotor yaw controls and led to substantial hysteresis effects with associated loss of energy capture. The wind turbine was then modified in December 1981 so that both the tail vane boom and the nacelle were independently supported by the yaw post with the help of ball bearings. The test and analysis results to be presented in the following are for this configuration involving negligible bearing friction for tail vane yawing and nacelle furling.

SECTION 2.0

TEST EQUIPMENT

2.1 EXPERIMENTAL TURBINE

The Tyson test equipment as of November 1981 was described in Ref. 2. Fig. 2.1 (a) shows a schematic side view of this turbine. The tail vane boom was supported by the nacelle 12 inch (0.30 m) downwind of the yaw post. Furling of the nacelle with respect to the tail boom involved the rotation of the nacelle about the yaw axis and a rotation of the tail boom about the furl pin. Since the furl pin rulon bearings were heavily loaded by the gravity moment of the tail vane and boom, furling of the nacelle caused a large frictional moment.

Fig. 2-1 (b) shows the side view of the turbine as modified in December 1981. Both nacelle and tail vane boom were supported by the yaw post. The tail vane boom was pinned to the yaw post and rotated together with it. The upper yaw post bearing was a self aligning Seal Master thrust ball bearing with low friction. The lower yaw post bearing was a Rulon bearing. Due to the large distance between upper and lower yaw post bearings (20 inch, 0.50 m), its load and friction moment was small. The nacelle was supported by the yaw post with the help of two self aligning Seal Master thrust ball bearings with low friction. The nacelle rotated (furled) relative to the yaw post and tail boom by 90 deg. Thus the yawing motion of the tail vane boom and the furling motion of the nacelle in relation to the boom were independent of each other, and they both had low friction moments. The original configuration of Fig. 2-1 (a) used a steel box as the main structure of the nacelle. The modified turbine, Fig. 2-1 (b), used instead a fiberglas composite bed plate. The rotor shaft supported by the bearing block and the shaft mounted gear box were same as before.

Fig. 2-2 shows the plan view of the modified experimental turbine of Fig. 2-1 (b). The hollow rotor shaft supported by the bearing block carried at its downwind end a slip ring unit with 16 slip rings (omitted in Fig. 2-1 (a) and (b)). The slip rings were connected through the hollow shaft to four strain gage bridges for measuring out-of-plane blade root bending moments, teeter motions and rotor torque. Other quantities measured were the rotor speed from a magnetic pickup signal, tail boom vertical and lateral bending, yaw post bending in two directions, axial bed plate acceleration, furl angle (the angle of the bed plate with respect to the yaw post), and generator current. The rotor axis was laterally off-set by 4.3 inch (0.11 m). The rotor overhang (distance to the yaw axis) was 32 inch (0.81 m). There were wedges between bearing block and bed plate seen in Fig. 2-1 (b) to adjust the uptilt angle of the rotor shaft.

The single phase Gould induction generator (not seen in Figures 2-1 and 2-2) was attached on top of the shaft mounted gearbox and was belt driven from the gear box. The transmission ratio between generator and rotor was normally 9.37 : 1. It could be varied by exchanging pulleys and V-belt. The break-out rotor torque was quite high, about 10 ft-lb (13 Nm). There were rather high losses from the shaft mounted gearing system and from the generator belt drive. The test results are given in terms of the measured rotor torque and rotor speed. No effort was made to evaluate the transmission and electrical losses.

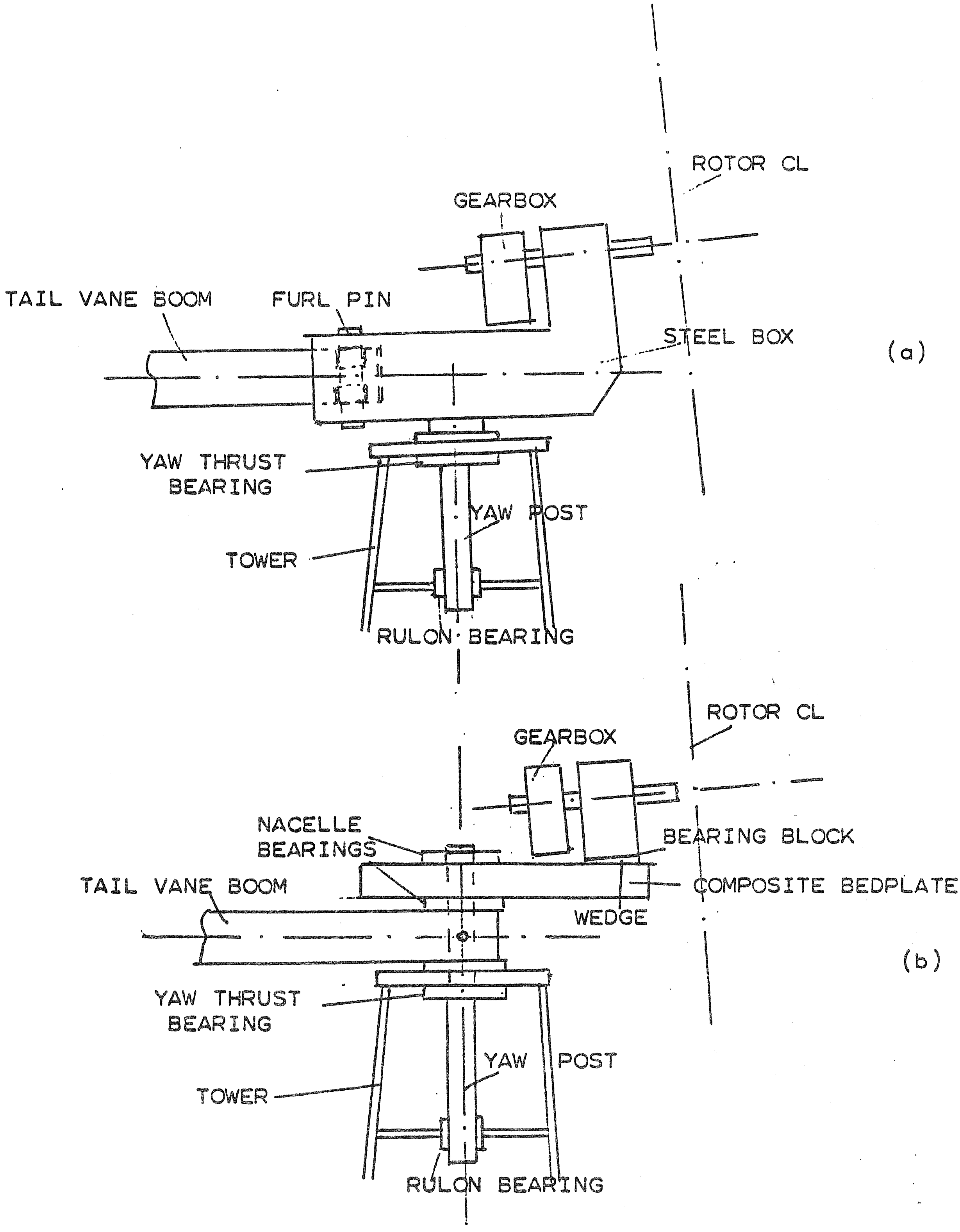


FIGURE 2-1. SIDE VIEW OF CONFIGURATION (a) USED 1980/81 AND (b) USED 1982

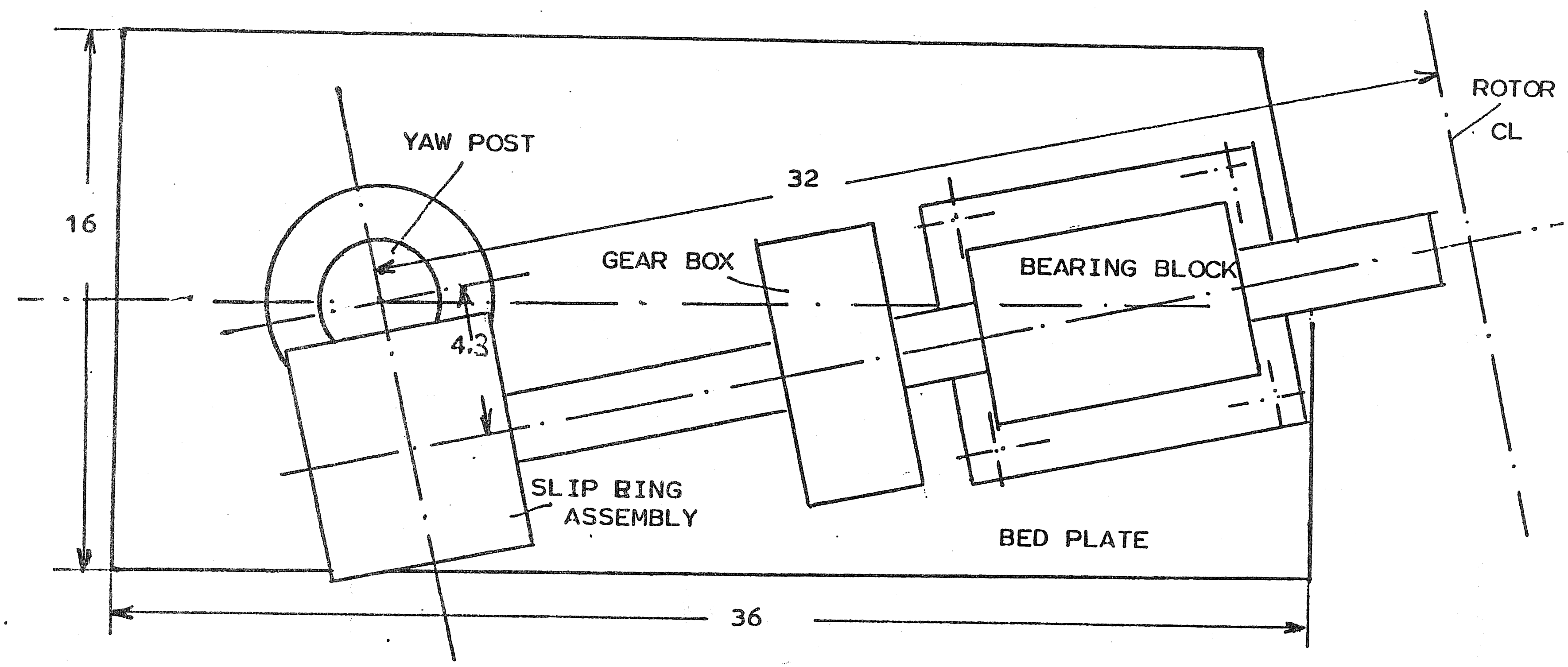


FIGURE 2-2. PLAN VIEW OF CONFIGURATION (b) .

2.2 PROOF-OF-CONCEPT TURBINE

In April 1983, the experimental turbine was changed to a duplicate of the Proof-of-Concept turbine tested at the Rocky Flats Wind Systems Test Center. Fig. 2-3 shows the plan view of this configuration with an integral gear-generator unit manufactured by U.S. Motors. This configuration was more compact than that shown in Fig. 2-2. The rotor overhang was reduced from 32 inch (0.81 m) to 27.5 inch (0.70 m). The transmission ratio of 9.21 was almost the same as before. The rotor axis lateral off-set from the yaw axis was also the same, 4.3 inch (0.11 m). The bed plate planform area was substantially smaller. The friction break-out torque was reduced from 10 ft-lb (13 Nm) to 2 ft-lb (2.7 Nm) and made the wind turbine start at lower wind speed.

The rotor shaft of the new configuration was too short to install a slipring unit so that measurements in the rotating system were no longer possible. However, during the preceding test period a good correlation had been established between axial acceleration and out-of-plane blade root bending moments. This allowed the estimation of the blade root bending moment from the measured axial acceleration. The main blade loads had their origin in the blade coning mode which was subjected to 2 P aerodynamic excitation. The blade coning mode amplitude was reflected in the axial 2 P nacelle acceleration. Teeter stop impacts were also clearly recognizeable in the axial acceleration trace so that this important limit condition could be established from the acceleration measurements. For the Rocky Flats tests, telemetering equipment was used for measuring quantities in the rotating system.

The generator was used at Rocky Flats in the three phase mode. At Tyson it was used in the single phase mode with a capacitance of 150 μF across one pair of phases. The three phases were conducted down to the instrument shed where capacitors were located. Tests had been conducted at U.S. Motors with the next larger unit of 50% increased rating (15 HP motor rating vs. 10 HP for the installed unit). It had been found that single phase operation was more efficient if the generator was driven in the sense opposite so that it would start as a motor.

Special starting capacitors were used for the motoring test. The Electromatic digital controller had a two second minimum delay between generator cut-out and cut-in. During these two seconds the generator could run as a motor driven by the grid but in a direction opposite so that it would turn when started as a motor from stand still.

During the 1982 tests the digital controller operated a mechanical contacter which was quite noisy. After installing the Proof-of-Concept turbine, an attempt was made to replace the mechanical contactor by a solid state switch. Due to high cut-in currents, the protective circuits blew several times. The solid state switch was then removed and the mechanical contactor was reinstated. At first the capacitance across one pair of phases was kept connected after generator cut-out. This produced high and erratic currents from self excitation. The capacitance was then wired such that it was cut out together with the generator using the third switch of the three pole contactor. Now the cut-in current transient varied little and stayed in the range of 50 to 60 amps peak current.

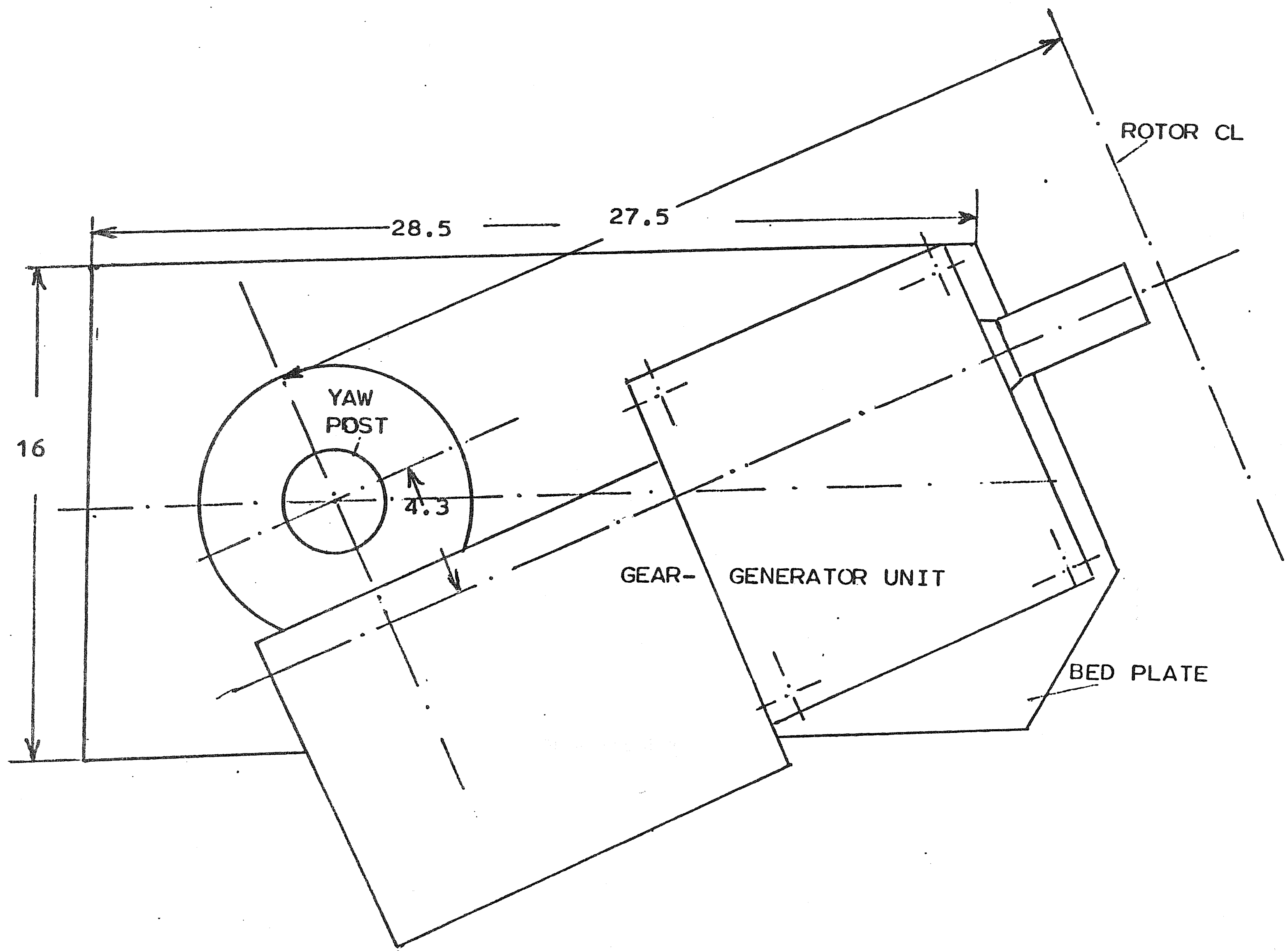


FIGURE 2-3. PLAN VIEW OF CONFIGURATION WITH GEAR-GENERATOR UNIT

Table 2-1 shows the results of the U.S. Motors tests for the single phase and for the three phase mode of operation, both for 240 volt. The actually measured currents and powers were reduced by a factor of 2/3 in order to apply to the units installed at Rocky Flats and at Tyson. The single phase data were taken in the mode where the generator is driven opposite to the direction it would start as a motor. The rotor power in Table 2-1 was determined with the assumption of a 0.1 KW friction loss and a 4% gearing loss. It is seen that in the middle power range three phase operation yields 90% generator efficiency, single phase operation only, 85% generator efficiency. For the tests with the Corps machine the data in Table 2-1 were used to determine electric power output from the measured current. This output checked well with an independent measurement from a power meter.

2.3 FURL CONTROL SYSTEM

Since the furl control system was the same both for the experimental and for the Proof-of-Concept turbines, there is no need to differentiate between these two configurations. Fig. 2-4 shows the plan view of the control system. Unfurl tension spring and actuator-damper unit were connected on the upwind side to the bed plate, on the downwind side to the tail boom. The electric actuator was not needed in normal operation and merely served to stop the rotor in case of an emergency or if the tower was to be lowered. Due to the lateral rotor axis off-set from the yaw axis, the rotor thrust produced a nacelle furling moment. As shown in Fig. 2-1 (b) the rotor axis was tilted upward. Since the rotor turned counter clockwise seen downwind, the driving rotor torque, due to this uptilt, contributed an unfurling moment. The uptilt angle of the rotor axis improved the rotor-tower clearance and allowed a small rotor overhang (distance to yaw axis) and a small nacelle moment of inertia. The power-on unfurling moment from torque compensated in part the difference between the high power-on and the low power-off rotor thrust moment at a given rotor speed. Without the rotor axis uptilt the power-off rotor speed would be excessive.

The main parameters of the furl control system were lateral rotor axis off-set, rotor axis uptilt angle, unfurl spring rate and unfurl spring preload, the kinematic relation between unfurl spring extension and furl angle, and the mechanical damper characteristics. Except for the first, all of these parameters were varied. Each of these changes required the lowering of the tower to gain access to the nacelle. Since lowering of the tower during a high wind period was inadvisable, the testing of each set of parameters required a separate high wind period of which there were at the site only a few per year except in non-usable thunder storms. This explains why researching the dynamic control characteristics of the turbine was time consuming. Furthermore, there was no mathematical model available for predicting the effects of the parameters on the dynamic control characteristics so that parameter changes had to be made by trial and error.

2.4 CHANGES TO EXPERIMENTAL WIND TURBINE

During the atmospheric testing, deficiencies in the test equipment were found and changes were made to correct the deficiencies. In the following the changes are discussed separately for the experimental turbine, for the

Table 2-1 U.S. Motors Generator Test Results

Single Phase Generator, 240 V with 140 μ f Across Two Phases

RPM	Amps	KW_{in}	KW_{out}	Effic.	PF	Rotor KW	Total Effic.
1868	50.6	12.0	8.7	0.72	0.75	12.6	0.69
1860	45	10.7	8.0	0.75	0.77	11.2	0.71
1853	39	9.5	7.3	0.77	0.80	10.0	0.73
1845	34	8.0	6.4	0.80	0.82	8.4	0.76
1838	29	6.7	5.6	0.82	0.84	7.1	0.79
1830	23	5.3	4.5	0.85	0.86	5.6	0.80
1824	18	4.1	3.5	0.85	0.85	4.3	0.81
1816	13	2.9	2.5	0.85	0.84	3.1	0.81
1808	7	1.5	1.1	0.73	0.66	1.7	0.65
1801	5	0.6	0.15	0.25	0.14	0.7	0.21

Three phase Generator, 240 V

RPM	Amps	KW_{in}	KW_{out}	Effic.	PF	Rotor KW	Total Effic.
1863	33	12.0	10.7	0.88	0.81	12.6	0.85
1855	29	10.7	9.5	0.89	0.81	11.2	0.85
1848	26	9.5	8.4	0.89	0.81	10.0	0.85
1841	25	8.0	7.1	0.90	0.79	8.4	0.85
1834	19	6.7	6.0	0.90	0.78	7.1	0.85
1827	16	5.3	4.8	0.90	0.75	5.6	0.85
1821	13	4.1	3.7	0.90	0.70	4.4	0.84
1814	11	2.9	2.5	0.87	0.55	3.1	0.81
1806	9	1.5	1.0	0.72	0.30	1.7	0.59
1802	8	0.6	0.2	0.33	0.06	0.7	0.28

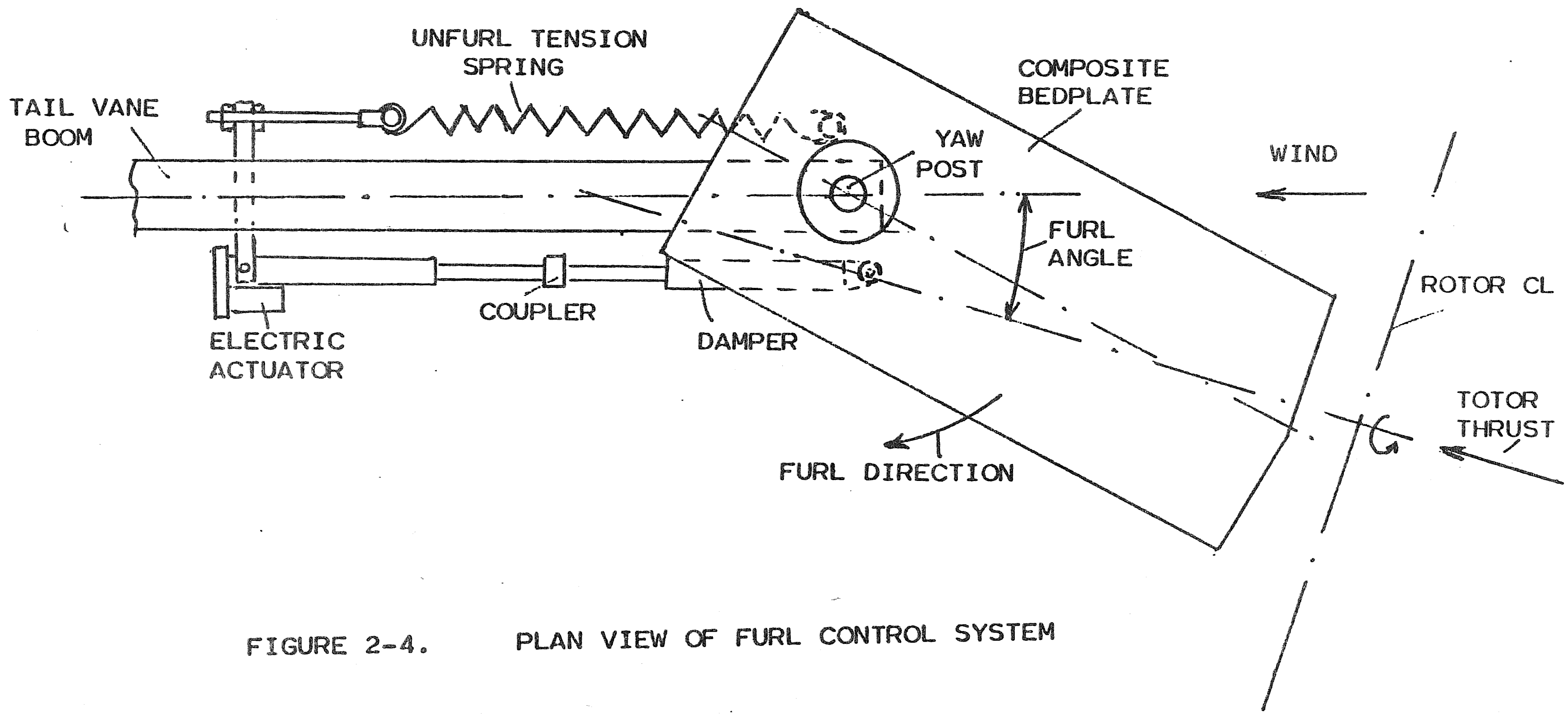


FIGURE 2-4. PLAN VIEW OF FURL CONTROL SYSTEM

Proof-of-Concept turbine and for the furl control system common to both. The changes are summarized for these three cases respectively in Tables 2-2, 2-3, 2-4.

2.4.1 Instrumentation Changes

In May 1982 a second anemometer was installed on top of the tail vane in order to obtain data on the effect of the rotor wake on the vane. In September 1982 a potentiometer pickup for the tail vane boom position relative to the tower was added. The yaw post rotation was transferred to a precision potentiometer with the help of double sprockets and a pinned drive belt. This instrument allowed to determine the total rotor yaw rate from nacelle furl and tail boom yaw measurements. Also in September 1982 the instrumentation was improved by changing the furl potentiometer to a strain gage flexure gliding in a spiral. This gave more reliable signals for the display instrument, for the oscillograph and for the computer.

2.4.2 Blade Change

In September 1982 the blades were modified as seen in Fig. 2.5. The added portion continued the leading and trailing edges to the station 23 inch (0.16 m) from rotor center. The twist increased within the added blade portion from 9 deg to 16 deg, the chord increased from 12 to 14 inch (0.30 to 0.36 m). The blade coning frequency was not affected by the addition and remained at 8.0 Hz. The effect of the modification was a rotor start-up at substantially lower wind speed so that the previously used starting motor could be eliminated.

2.4.3 Tail Vane and Tail Boom Changes

Also in September 1982 a new enlarged tail vane and a longer tail boom were installed as seen in Fig. 2.6. The tail vane area was increased by a factor of 1.5 from 16 sq ft to 24 sq ft (1.5 m^2 to 2.25 m^2). The tail vane height was increased by a factor of 1.2 from 120 inch to 144 inch (3.0 to 3.6 m). The mean tail vane chord was increased by a factor of 1.25 from 19 inch to 24 inch (0.48 m to 0.61 m). The boom length between yaw axis and vane trailing edge increased by a factor of 1.12 from 192 inch to 216 inch (4.9 to 5.5 m). There were three reasons for the change. First, it had been observed that at low wind speed power-on, vane yawing was erratic which was interpreted as caused by the onset of vane stall. In other words, at low wind speed, the tail vane could not resist the furling moment of the rotor thrust. This condition was corrected by the larger tail. Second, the vertical tail natural frequency was rather close to the operational rotor speed so that quite large 1 P vertical tail excitation occurred. The modification lowered the natural vertical tail frequency from 180 CPM to about 140 CPM so that at the rotor operational speed of 195 RPM no visible 1 P vertical tail vibrations could be observed. Third, strong 4 P torque oscillations together with 4 P nacelle yaw oscillations had been observed at high power output. This condition was interpreted as a combined blade in-plane and lateral tail boom mode coupled through the rotor shaft uptilt angle. The modified tail boom had a substantially lower lateral natural frequency which no longer coupled with the in-plane blade mode.

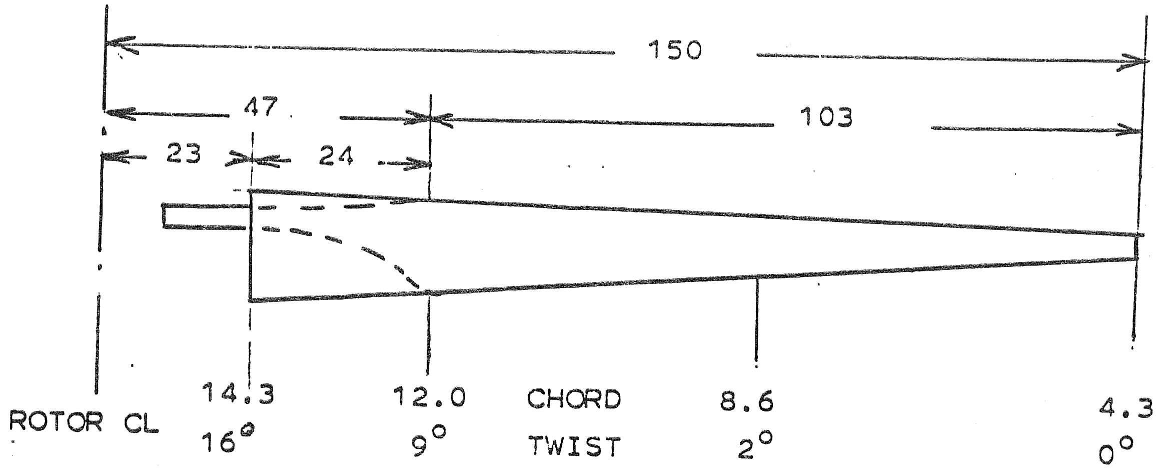


FIGURE 2-5. BLADE PLANFORM BEFORE (DASHED LINES) AND AFTER MODIFICATION, DIMENSIONS IN INCHES

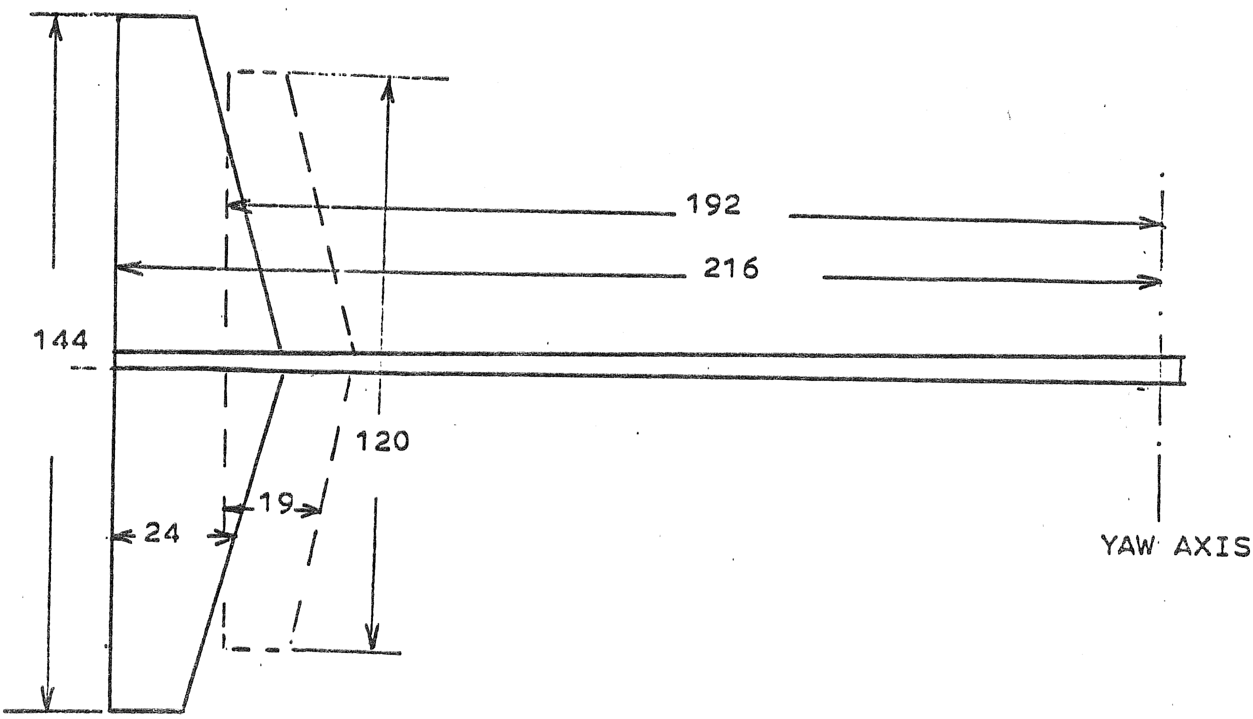


FIGURE 2-6. TAIL PLANFORM AND TAIL BOOM BEFORE (DASHED LINES) AND AFTER MODIFICATION, DIMENSIONS IN INCHES

2.4.4 Transmission Ratio Increase

In November 1982 the transmission ratio between induction generator and rotor was changed from 9.37 : 1 to 12.5 : 1 by using a different size pulley in the belt drive. The purpose of this change was to study the stall characteristics of the rotor.

2.5 CHANGES TO THE PROOF-OF-CONCEPT TURBINE

All the changes to the experimental turbine discussed above (except for the change in transmission ratio) were carried over to the Proof-of-Concept turbines. The additional changes to this turbine made in 1983 to 1985 are discussed in the following subsections and summarized in Table 2-3. None of these changes were made to date to the Rocky Flats PCP turbine.

2.5.1 Aluminum Disk on Shaft

In April 1983 a 9 inch (0.23 m) diameter aluminum disk was mounted on the rotor shaft as seen in Fig. 2-7. The disk had a slight press fit and was also secured to the shaft by a key not shown in Fig. 2-7. A spacer was inserted between the disk and the cone of the taper lock that connected the hub to the shaft. The disk was used to support the teeter tension springs and later also the teeter compression springs and the friction pad of the rotor brake added in May 1985.

2.5.2 Tension Teeter Springs

The rotor hub was modified in May of 1983 by the addition of aluminum angles as seen in Fig. 2-8. The angles were bolted to the hub box by the four bolts holding the teeter bearing block inside the hub box as seen in the view along the rotor axis in the lower part of Fig. 2-8. Tension teeter springs were then mounted between the aluminum disk and the angles as shown in Fig. 2-8. Several sets of tension teeter springs were tested. Most of the tests were conducted with a set of teeter springs that gave an elastic restraint of 27 ft-lb/deg of out-of-plane blade deflection (37 Nm/deg). A set of weaker springs had little effect on the control characteristics. Stronger springs caused 1 P nacelle excitation, presumably because of unequal spring preload which could not be removed since the spring lengths were not adjustable.

2.5.3 Emergency Shut-Down System

Fig. 2-9 shows a block diagram of the electrical system. The upper portion of the diagram contains the elements that were installed in September 1981, namely circuit breaker, power contactor, digital controller, magnetic pick-up. The remaining components represent an emergency shut-down system which was designed and installed in March of 1984 in preparation for unattended turbine operation. The latching relay had two functions; first, to operate the power contactor, second to power the furl actuator in response to one of four emergency signals indicating power loss, overspeed, overvoltage and excessive vibrations. The furl actuator could be switched either to the manual or to the automatic mode when the furl signal would be given by the latching relay. When the actuator was in the automatic mode, any of the four emergency signals would result in generator disconnect and in rotor furling.

The power failure relay operated an adjustable timer which was set to about one second delay. When the power returned within one second, the latching relay would not trip, the generator would remain connected and the rotor would remain unfurled. The overspeed relay would receive its closing signal from the frequency-to-voltage converter which was receiving pulses with twice generator RPM from a magnetic pick-up. This pick-up was separate from that feeding the digital controller. The frequency-to-voltage converter also fed voltage proportional to rotor speed to the Vishay signal conditioner and from there to the RPM display instrument, to the oscillograph and to the computer. The latching relay remained tripped until an unlatching switch was operated. During unattended operation the circuit breaker tripped once because of generator overload and the machine was found in furled position with the rotor standing still. The overspeed shut-down was set to 10% overspeed. Actually, the functioning of the overspeed and the power failure relays and even the functioning of the electric actuator were not critical since the mechanical furl control system would prevent a dangerous overspeed. However, at power-off rotor speed in the vicinity of 20 mph (9 m/s) wind velocity the blade coning frequency was in near resonance with 2 P aerodynamic excitation. Several days of power-off operation might have reduced turbine life. The Rocky Flats PCP turbine had no emergency shut-down provisions and was not meant to operate unattended.

2.5.4 Vibration Shut-Down Switch

The Select Controls vibration shut-down switch was installed in October 1984. When positioned parallel to the rotor axis, it was closed below 1.2 g and open above 1.2 g. It was found that during generator cut-in a very brief acceleration peak occurred which occasionally tripped the acceleration switch. The switch had to be installed at an angle to the axial direction to make it less sensitive. Though normally the axial accelerations were not over ± 0.5 g, the acceleration switch had to be desensitized to about 1.8 g in order to reliably prevent tripping from generator cut-in.

2.5.5 Compression Teeter Springs

In order to test higher teeter spring rates, a set of adjustable compression springs was added in May 1985 (Fig. 2-10). Adjustable tension rods were supported by the aluminum disk and carried on the upwind side of the hub box the compression springs which gave an elastic restraint of 35 ft-lb/deg of out-of-plane blade deflection (47 Nm/deg). The tension teeter springs were left installed but their unequal preload could now be compensated by the adjustable preloads of the compression springs. The combined elastic restraint was 62 ft-lb/deg (84 Nm/deg). The teeter bearings and the rotor shaft were quite highly loaded by the combined teeter springs which should serve merely research purposes but should not be used for long time operation. The fatigue strength of the shaft and the allowable radial loads of the Rulon teeter bearings are adequate for long term operation of either of the tension or compression teeter springs alone.

2.5.6 Rotor Brake

After the Corps terminated the support in December 1984, continued testing depended on changing the wind turbine to a configuration for which no experienced help would be needed when raising and lowering the tower. This

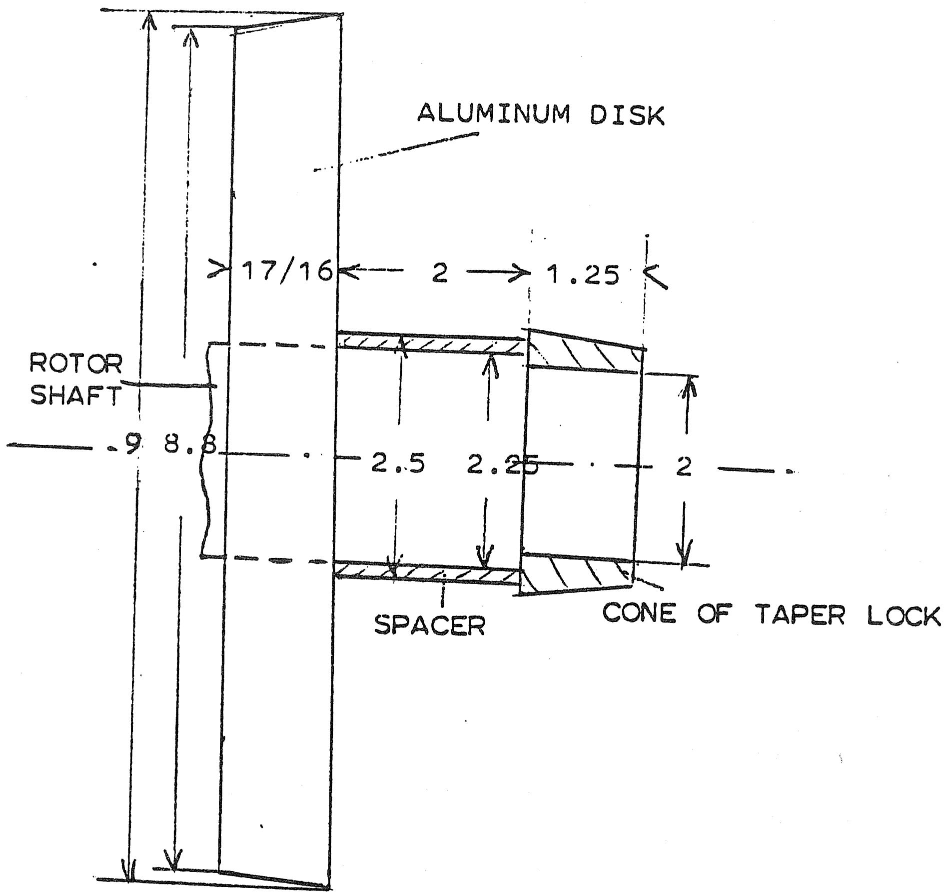


FIGURE 2-7. TEETER SPRINGS SUPPORTING DISK ASSEMBLY, 1/2 SCALE

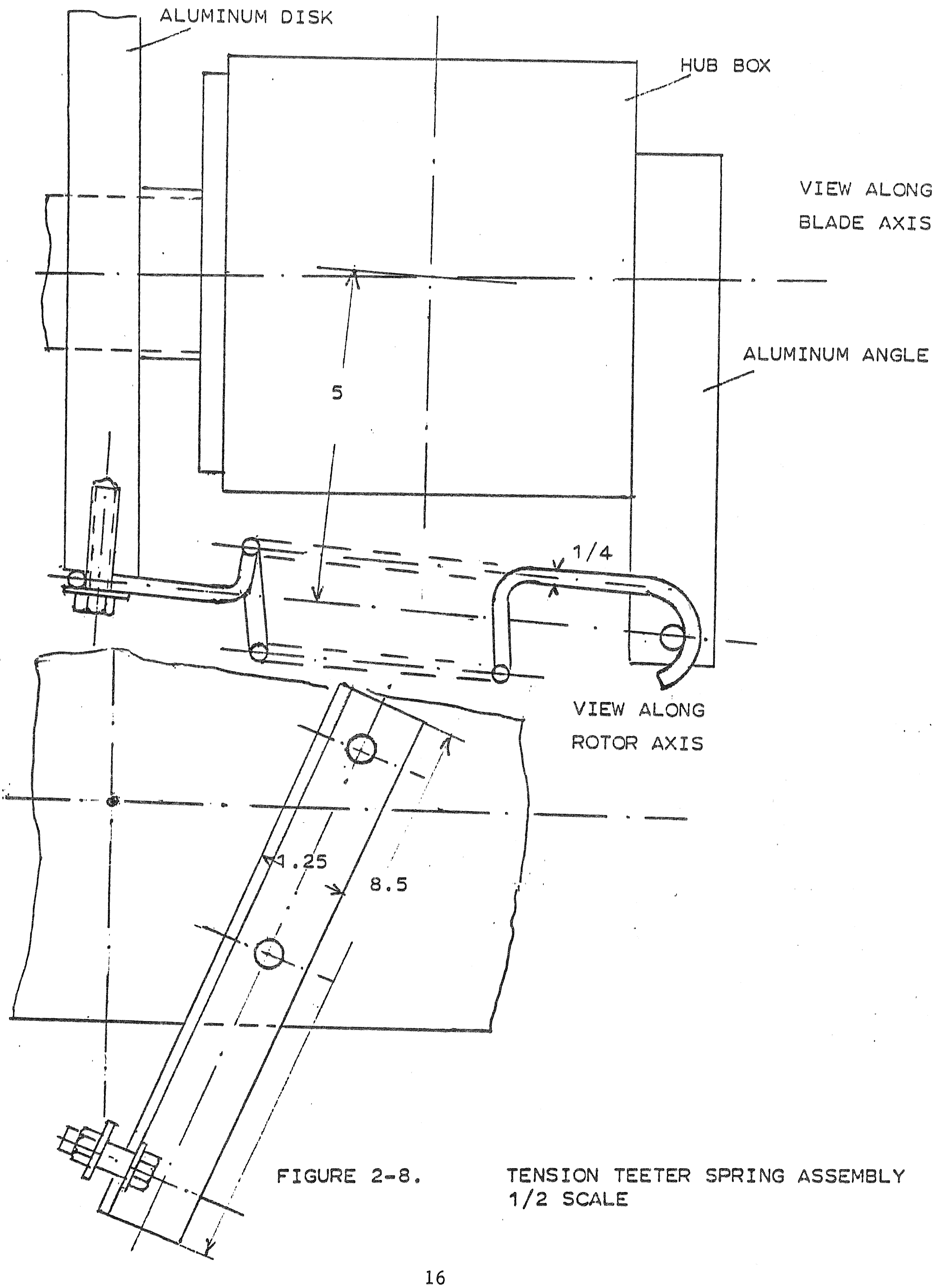


FIGURE 2-8. TENSION TEETER SPRING ASSEMBLY
1/2 SCALE

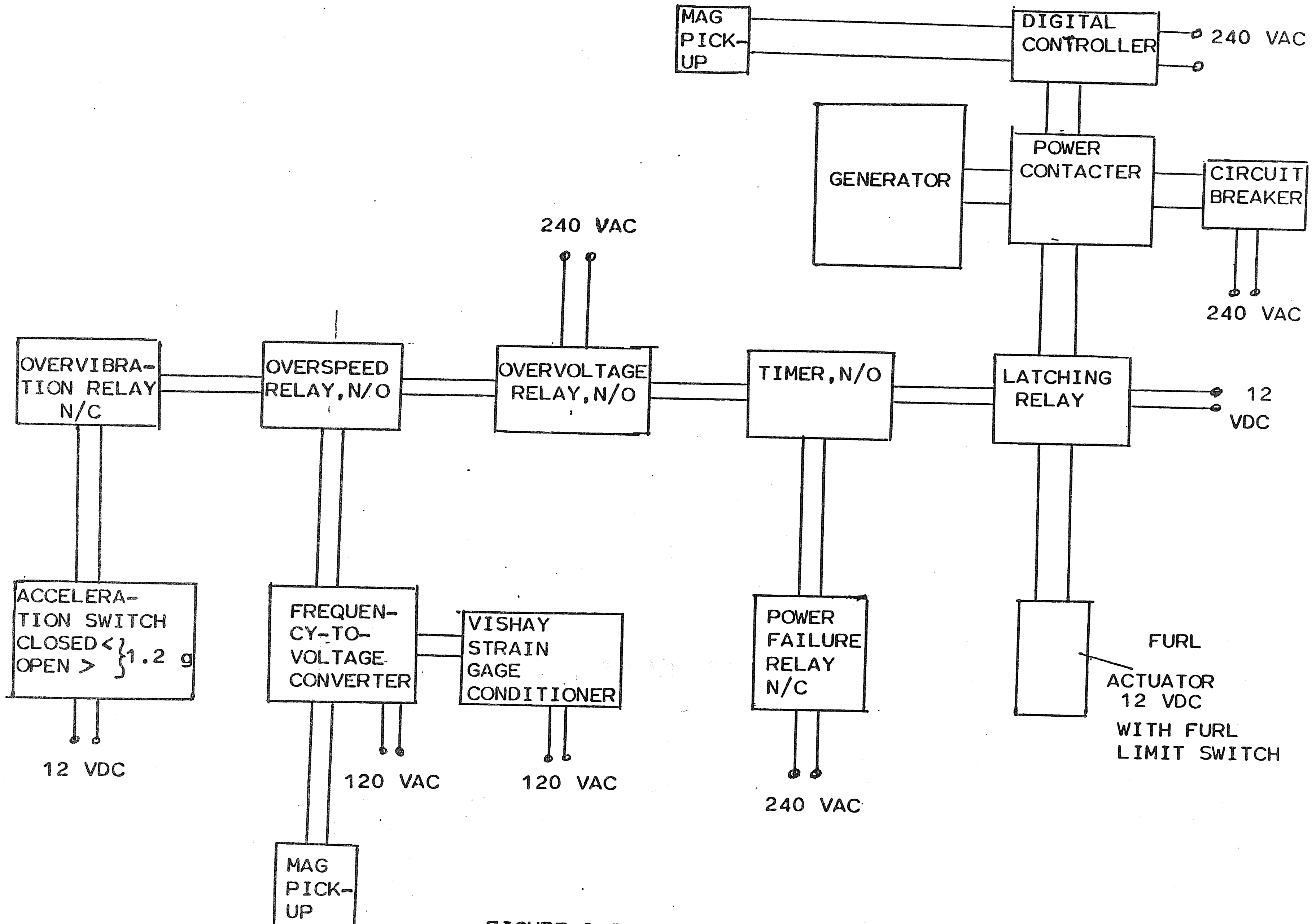


FIGURE 2-9. SCHEMATIC OF ELECTRIC CONTROL SYSTEM

required the design, manufacturing and testing of a rotor brake to be able to raise and lower the tower in high winds. The brake was installed in May 1985 and tested in September 1985. As seen in Fig. 2-11, the brake pad was pressed by an electric actuator against the downwind side of the aluminum disk of Fig. 2-7. The walking beam carrying the brake pad was supported by one of the bolts that tie the gear-generator unit to the bed plate. The electric 12 VDC actuator had an adjustable slip clutch by which the force normal to the brake pad could be limited. The brake provided a rotor friction torque of 20 ft-lb (27 Nm) which compared to 2 ft-lb (2.7 Nm) of break-out friction torque without the brake. The brake made it possible to raise and lower the tower in winds up to 20 mph (9 m/s).

2.6 CHANGES IN FURL CONTROL SYSTEM

The changes are summarized in Table 2-4.

2.6.1 Actuator Guide Rods

The modification to the electric furl actuator seen in Fig. 2-12 was made in January 1982. It had been observed that the actuator piston occasionally rotated about its axis by several degrees which changed the piston axial position. Guide rods were then added held by a clamp around the coupler and sliding in holes through another clamp around the actuator body. Due to the ball screw in the actuator, the torsional moment tending to rotate the piston was small and the guide rods were capable to prevent such rotation. The guide rod clamps had to be adjusted such that even with fully extended actuator the guide rods were still inside the guiding holes. Due to an adjustment error of the clamps, the guide rods were pulled out of the guide holes during unfurling in April 1984 resulting in a blockage of the actuator when trying to furl. The rotor kept turning for 24 hours in winds between 5 and 15 mph so that a crane with bucket had to be rented to furl the rotor externally and stop it before lowering the tower.

2.6.2 Rigid Damper-Actuator Coupling

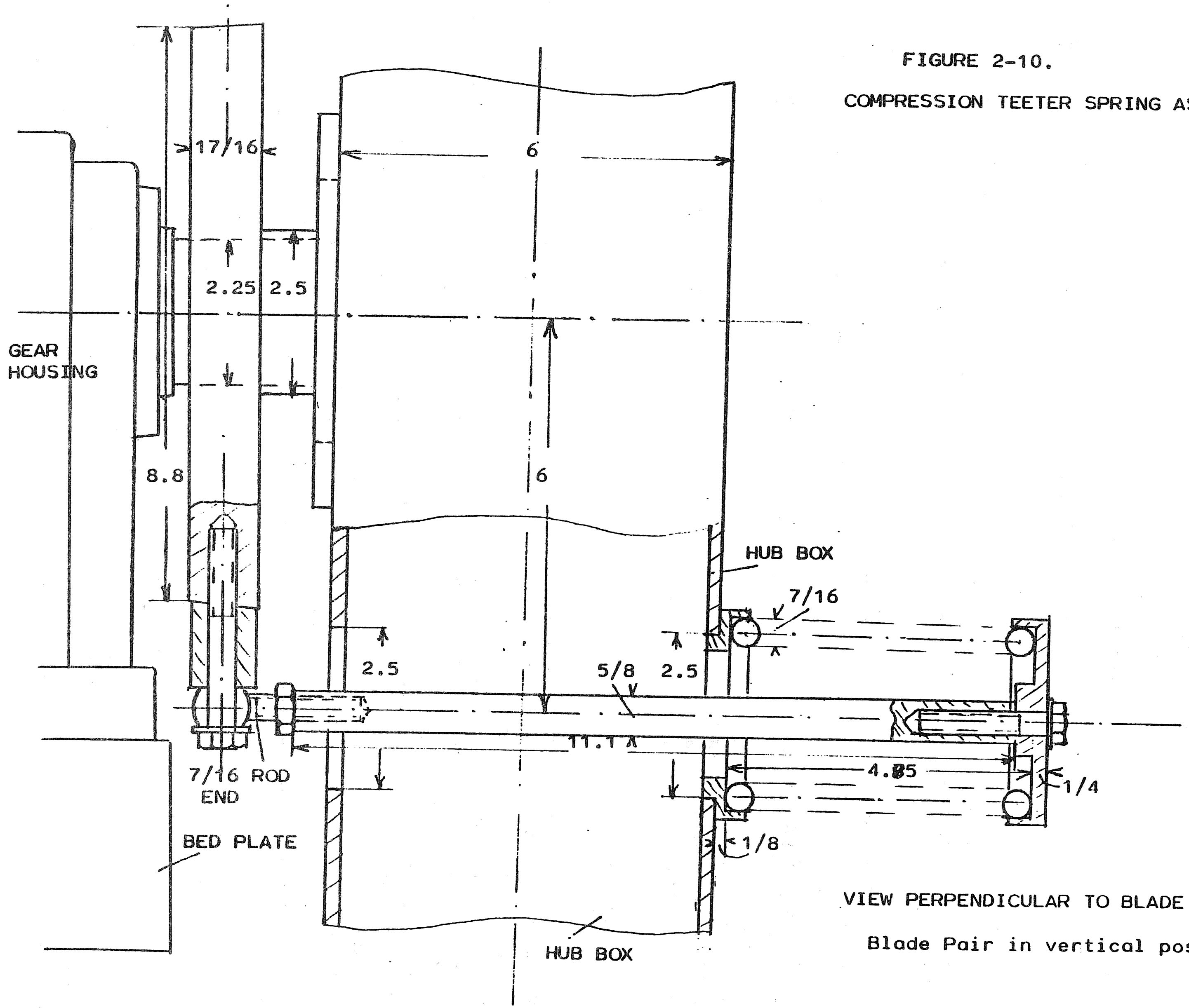
Originally, the actuator piston and the damper rods were connected by pin and clevis as shown in the assembly drawing of Ref. 2. In March 1982 this connection was replaced by a rigid coupler which prevented jack knifing of the unit when the compression from the damper during furling exceeded the tension force from the unfurl spring.

2.6.3 Enidine Damper

The control dynamics of the furl system was found to depend critically on the furl damper, and numerous damper versions were tested. The original damper designed and built by Washington University Technology Associates (WUTA) had an overflow reservoir that was in part filled with compressed air. The damper force at a given damper rate could be varied by changing a valve setting, but the force depended also on the reservoir air pressure. The damper had linear characteristics with the force proportional to the rate. Temperature had a large effect on the damper. For example, the damper rate under 40 lb force changed from 0.5 inch/s at 20 deg F to 1.0 inch/s at 70 deg F.

FIGURE 2-10.

COMPRESSION TEETER SPRING ASSEMBLY



VIEW PERPENDICULAR TO BLADE AXIS

Blade Pair in vertical position

In June of 1982 a damper manufacturered by the Enidine Corporation was installed. It had a rubber sponge material in the cylinder that could soak up the hydraulic fluid displaced by the damper rod when it was retracted into the cylinder. The damper force was proportional to the square of the rate. The force was not adjustable. For a change of damper characteristics an insert containing the fluid passages had to be replaced which required drainage, disassembly, reassembly and refilling of the damper. The original Enidine damper had excessive break-out friction. The damper was returned to the manufacturer and the seals were replaced by O-rings which reduced the break-out friction with the piston in the middle range to about 10 lb (44 N). The modified Enidine damper was reinstalled at Tyson in December 1982. The damper, due to the special silicon based hydraulic fluid, retained its characteristics nearly independent of ambient temperature. The first insert produced at 50 lb (222 N) damper force a damper rate of 1.0 inch/s (0.025 m/s). The same type of insert and the same type of O-rings were used later for the two Proof-of-Concept turbines. In the course of time, three further inserts were acquired and tested at Tyson with damper rates under 50 lb (222 N) force of 0.50 inch/s (0.013 m/s), 0.70 inch/s (0.018 m/s) and 1.30 inch/s (0.033 m/s).

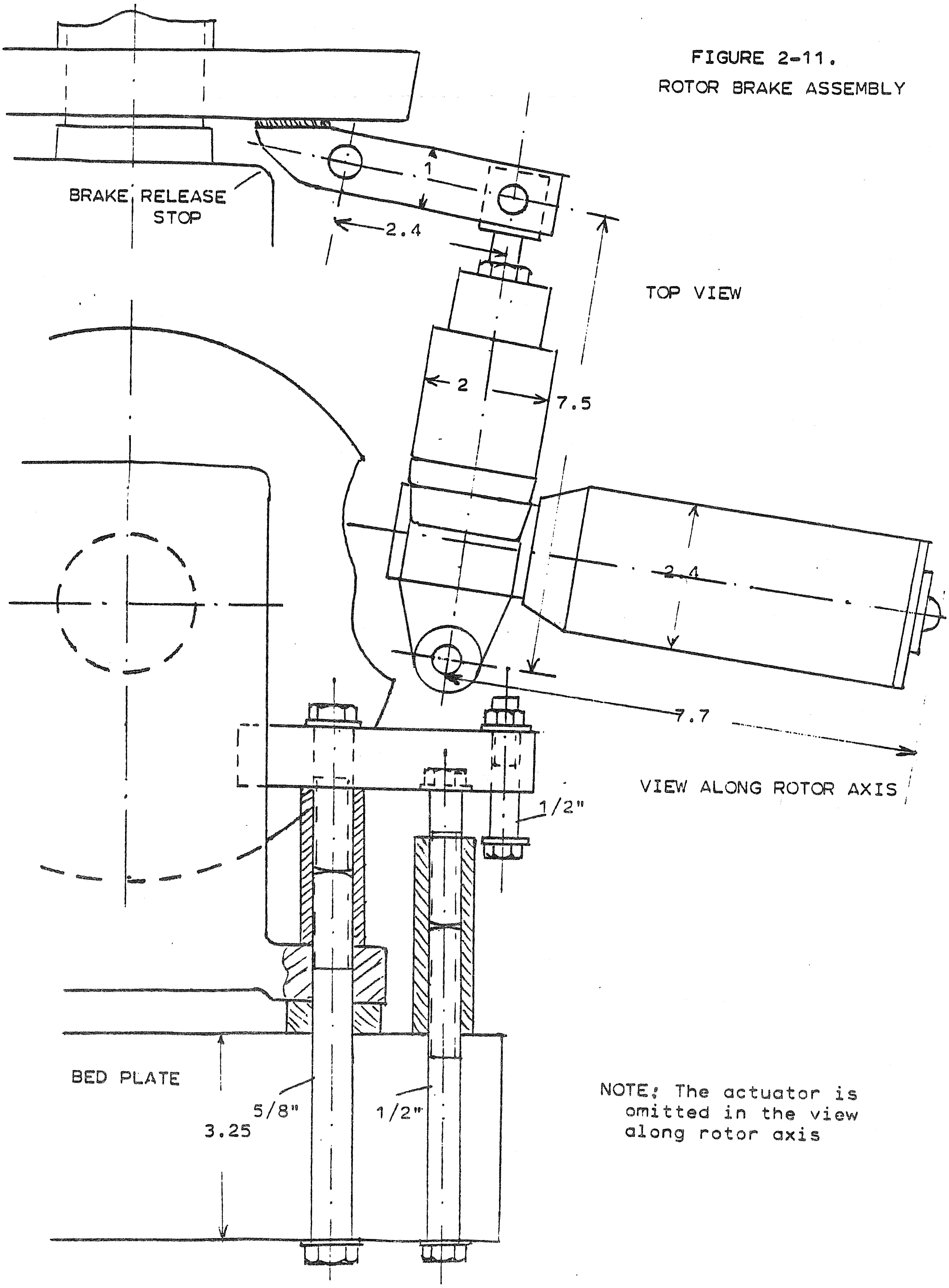
The damper was taken off the turbine once and stored for several months in partially compressed position. After reinstalling the damper, it was found to have erratic damping characteristics. The manufacturer was called and gave the belated information that the damper should only be stored in extended position so not to damage the sponge rubber insert. After storing the damper for a few weeks in extended position and after topping the damper fluid, it functioned properly again. Fortunately, when installed in the turbine, the damper was always in its extended or near extended position except during relatively short high wind periods.

2.6.4 Elastic Unfurl Stop

Fig. 2-12 (a) shows the normal operating condition of the electric actuator with extended piston. Originally, there was no unfurl stop between bed plate and tail vane boom so that the unfurl limit was given by the fully extended damper. This was true also for the Rocky Flats PCP turbine. Unfurl impact forces went into the damper and into the clevis attaching the damper to the bed plate. As a consequence of these impact forces (and of a faulty clevis), the connection of the damper to the bed plate failed. The nacelle then performed oscillations between the furl and unfurl limits (87 deg to -15 deg furl angle) with a period of 5.5 seconds and with a furl and unfurl rate of 50 deg/s. The blade loads during these oscillations were not higher than before the failure. Since the actuator was disconnected, the rotor had to be stopped manually after the winds dropped below cut-in with the help of the crane and bucket.

The Tyson Proof-of-Concept machine had a much more massive damper clevis taken from the preceding experimental turbine. It could accept the unfurl impact loads. A stronger damper clevis was also installed at Rocky Flats. As a consequence of the clevis failure, both machines, that at Rocky Flats and that at Tyson, received in April 1983 an elastic unfurl stop between nacelle and tail vane boom so that the unfurl impact loads were absorbed by this stop rather than by the damper. Another important advantage of the unfurl stop was that now the damper piston could be placed in the unfurled position about 1.5

FIGURE 2-11.
ROTOR BRAKE ASSEMBLY



NOTE: The actuator is omitted in the view along rotor axis

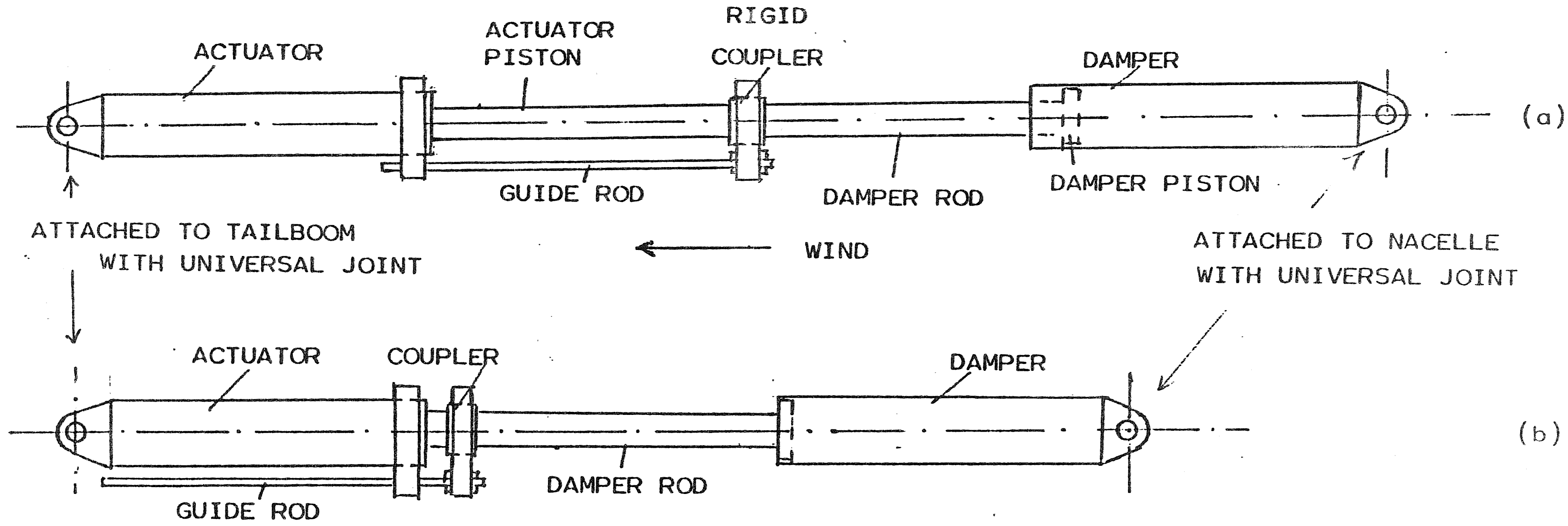


FIGURE 2-12. ACTUATOR-DAMPER UNIT (a) OPERATING CONDITION
(b) EMERGENCY FURLED

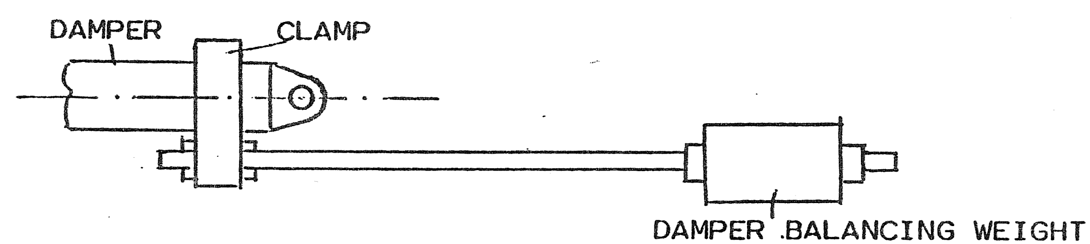


FIGURE 2-13. CLAMPED-ON DAMPER BALANCING WEIGHT

inch (0.038 m) away from its fully extended location as seen in Fig. 2-12 (a). This substantially reduced the damper break-out force.

During emergency furling with the actuator the damper rod had to be pulled out all the way as seen in Fig. 2-12 (b). The nacelle was furled to the furl stop, positioned at 87 deg furl angle. This brought the rotor to a stand still up to 30 mph (13.4 m/s) wind speed. Whether the rotor will stand still at much higher wind speed when furled to this limit is not known.

2.6.5 Damper Balancing Weight

Due to the damper gravity bending moment, the damper break-out friction increased substantially when the piston came close to the fully extended position. In May 1983 the damper for the Tyson turbine, but not for the Rocky Flats machine, was balanced by a device shown in Fig. 2-13 balancing one half of the damper gravity moment about the nacelle attachment bearing. This measure together with placing the piston in the unfurled position 1.5 inch away from fully extended, reduced the damper break-out force in the unfurled condition from 100 lb (445 N) to 20 lb (89 N).

2.6.6 Furl Limit Switch

In April 1984 a furl limit switch was added which interrupted actuator power at contact with the furl stop. This limit switch was part of the automatic emergency shut-down equipment installed at Tyson for use during unattended wind turbine operation. The actuator had a slip clutch which allowed a few seconds of operation against the furl stop. However, powering the actuator for an extended period beyond stop contact would have burnt out the slip clutch.

2.6.7 Stiffer Unfurl Spring

In May 1984 the unfurl spring with 5.7 lb/inch (1005 N/m) was exchanged for one with 7.1 lb/inch (1250 N/m). The advantage of the stiffer spring was that spring preload adjustments were easier to make. Previously, the rod end had to be reversed for some of the tested high preload conditions which meant disconneting the spring. With the stiffer spring the rod end could remain in place for all spring preload adjustments.

TABLE 2-2. Changes to Experimental Turbine

Date	Item	Effect
5-82	Anemometer on Tail Vane	Rotor Wake Measurement
9-82	Tail Boom Position Indicator	Total Rotor Yaw Rate
9-82	Furl Strain Gage Indicator	Greater Reliability
9-82	Blade Airfoil Extension	Earlier Start-up
9-82	Larger Tail Vane, Larger Boom	Erratic Yawing Avoided
12-82	Increased Transmission Ratio	Blade Stall Study

TABLE 2-3. Changes to Proof-of-Concept Turbine
(None Made to Rocky Flats PCP Turbine)

Date	Item	Effect
4-83	Aluminum Disk on Shaft	Supported Teeter Springs
5-83	Tension Teeter Springs	Delayed Furling
3-84	Emergency Shut-Down System	Unattended Operation
10-84	Vibration Shut-Down Switch	Unattended Operation
5-85	Compression Teeter Springs	Delayed Furling
5-85	Rotor Brake	Tower Tilting in Wind

TABLE 2-4. Changes to Furl Control System

Date	Item	Effect
1-82	* Actuator Guide Rods	No Actuator Piston Rotation
3-82	* Rigid Damper Coupling	No Jack Knifing
6-82	* Enidine Damper	Greater Reliability
12-82	* Enidine damper O-Rings	Lower Break-Out Friction
4-83	* Elastic Unfurl Stop	No damper Impact Loads
5-83	Damper Balance Weight	Lower Break-Out Friction
4-84	Furl Limit Switch	Unattended Operation
5-84	Stiffer Unfurl Spring	Delayed Furling

* Incorporated in Rocky Flats PCP Turbine

SECTION 3.0

ANALYSIS DATA

Though analysis was not a requirement of the subcontract, some analytic data will be given here since they serve to better understand the experimental data. The analysis was limited to the determination of yaw trim conditions and to the prediction of rotor yaw damping and rotor yaw position stability. A complete yaw control dynamics analysis has not as yet been performed. The analysis was based on a model developed in Ref. 3. A slow prescribed rotor yaw oscillation was analyzed and the rotor yaw damping was determined from the response. The model of Ref. 3 also allowed the determination of hub forces and moments as a function of steady yaw angle, thus making a yaw trim analysis possible. The yaw trim analysis results presented in the following were taken from Ref. 4, the rotor yaw damping results from Ref. 5.

3.1 YAW TRIM ANALYSIS RESULTS

The equivalent rotor thrust was defined by

$$a\ T_{EQ} = (T\ a + H\ S - Q\eta) \tag{3-1}$$

The aerodynamic furling moment was T_{EQ} a. Without the negative term $- Q\eta$ from rotor axis uptilt angle η, one would need for torque limitation a substantially higher unfurl spring moment. This would drive up the power-off rotor speed. For the selected value of $\eta = 7$ deg the power loss from axis uptilt angle is small. If teeter springs were used, a fourth term was needed in the expression for T_{EQ}. The following trim analysis results were obtained without the teeter spring term. In non-dimensional form and using the parameters of the experimental turbine Eq. 3-1 reads

$$c_{TEQ} = c_T + 7.8\ c_H - 5.0\ c_Q \tag{3.2}$$

Aerodynamically more meaningful are these coefficients when divided by the blade solidity ratio.

Fig. 3-1 taken from Ref. 4 shows c_{TEQ}/σ vs. rotor furl angle for 200 RPM. The two sets of solid lines represent constant torque coefficient c_Q/σ and constant speed ratio $V/\Omega R$. Estimates of the nacelle yawing moment without rotor were included in the determination of the horizontal force coefficient c_H. The dashed line represents the unfurling moment of the tension spring in terms of c_{TEQ}/σ. The reduction in spring moment for furl angles over 60 deg was caused by the spring kinematics since the center-line of the tension spring had at high furl angles a smaller moment arm with respect to the furl axis than at medium furl angles. Trim balance was obtained when the aerodynamic furling moment was equal to the spring unfurling moment. Following the dashed line from left to right, it is seen that increasing wind speed caused increasing furl trim angles. Up to 40 deg furl angle and up to a speed ratio of 0.17 the rotor torque remained nearly constant. For higher wind speed the rotor torque diminished and reached zero at $V/\Omega R = 0.31$. Due

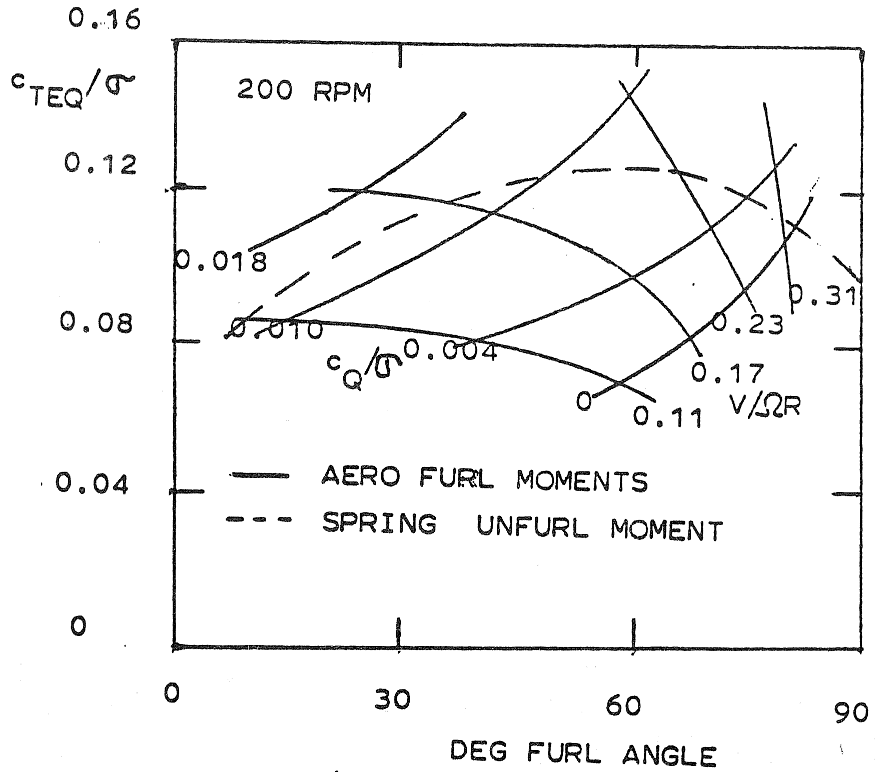


FIGURE 3-1. EQUIVALENT ROTOR THRUST COEFFICIENT VS. FURL ANGLE

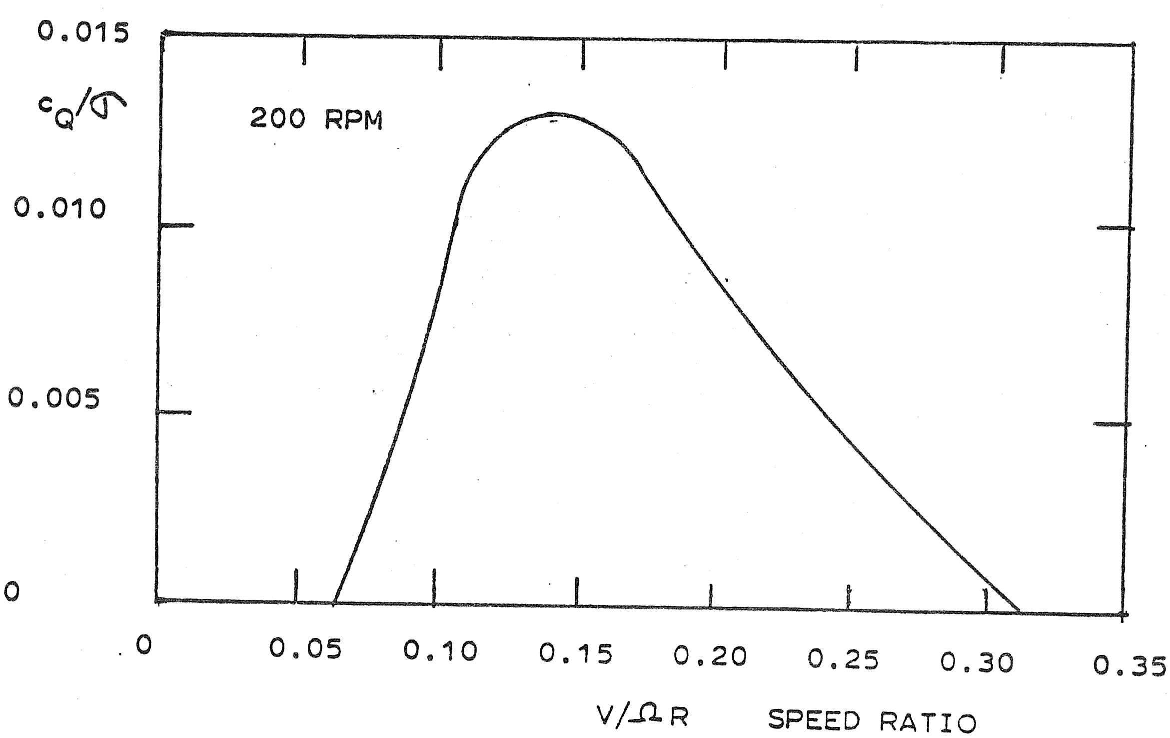


FIGURE 3-2. TORQUE COEFFICIENT VS. SPEED RATIO

to the steep negative slopes of the lines of constant $V/\Omega R$ at high furl angles, the trim position at a given wind speed was always stable despite of the negative spring moment slope. This means that at a given wind speed an increase in furl angle was always associated with an unfurling restoring moment.

Fig. 3-2 shows at 200 RPM the trim values of the torque coefficient c_Q/σ vs. the speed ratio $V/\Omega R$. Generator cut-in speed was $V/\Omega R = 0.07$ and cut-out speed was $V/\Omega R = 0.31$. Fig. 3-3 shows power-off RPM and furl angle vs. wind speed from analysis (solid line) and from a test on April 3, 1982 (crosses). The test results will be discussed later. Maximum power-off RPM was reached at 20 mph (9 m/s) wind speed. At 50 mph (22 m/s) the rotor speed was down again to 200 RPM. Beyond 50 mph (22 m/s) the RPM should drop below 200.

It should be noted that the digital controller cut power out when the generator speed was below synchronous and cut power in when the speed was above synchronous. The grid disconnect at $V/\Omega R = 0.31$ or 50 mph (22 m/s) wind speed was, therefore, automatic without the necessity of any wind speed signal, the grid connection at $V/\Omega R = 0.07$ or 11 mph (5 m/s) was also automatic and needed no wind speed measurement. Due to fluctuating wind speed it was unavoidable that in the vicinity of the cut-in and cut-out velocities frequent on and off generator switches occurred. Due to the two second delay built into the digital controller this was acceptable though some motoring losses were involved.

3.2 ANALYSIS RESULTS ON ROTOR YAW DAMPING AND YAW POSITION STABILITY

The analytical model developed in Ref. 3 was used in Ref. 5 to determine rotor yaw damping and rotor yaw position stability as a function of teeter spring stiffness. The latter was characterized by the parameter P which represents at a given rotor speed the teeter natural frequency with teeter springs over that without teeter springs, omitting aerodynamic effects. The paramter P - 1 is nearly proportional to the teeter spring stiffness. For given teeter springs, P is a function of rotor speed and becomes larger at lower rotor speed. The total teeter stiffness (without aerodynamic effects) is the sum of the spring stiffness and the centrifugal stiffness. At lower rotor speed the latter becomes less in comparison to the teeter spring stiffness so that the relative spring effect becomes larger.

The two upper graphs of Fig. 3-4 show the results of the rotor yaw damping analysis for $\delta_3 = 0$ and 67 deg. The main parameters affecting the results are thrust coefficient c_T/σ, angular rotor speed Ω, Lock number γ, rotor diameter D and rotor overhang S. It was found that the rotor yaw damping moment was nearly proportional to the yaw rate. Thus only one yaw rate of 0.5 rad/s is shown in Fig. 3-4. The remaining parameters are listed on top of this Figure. The yaw damping moment was assumed to be positive when directed opposite to the yaw rate. The mean spring hub moment is represented by the top dashed lines of the upper two graphs and contributed to positive yaw damping. The yaw rate produced a tilting of the rotor tip path plane with associated thrust and torque vector tilting and with an associated horizontal side force. For an upwind rotor all these effects produced a negative rotor yaw damping. The aerodynamic effects due to the nacelle without rotor were omitted for Fig. 3-4. It is seen that without teeter

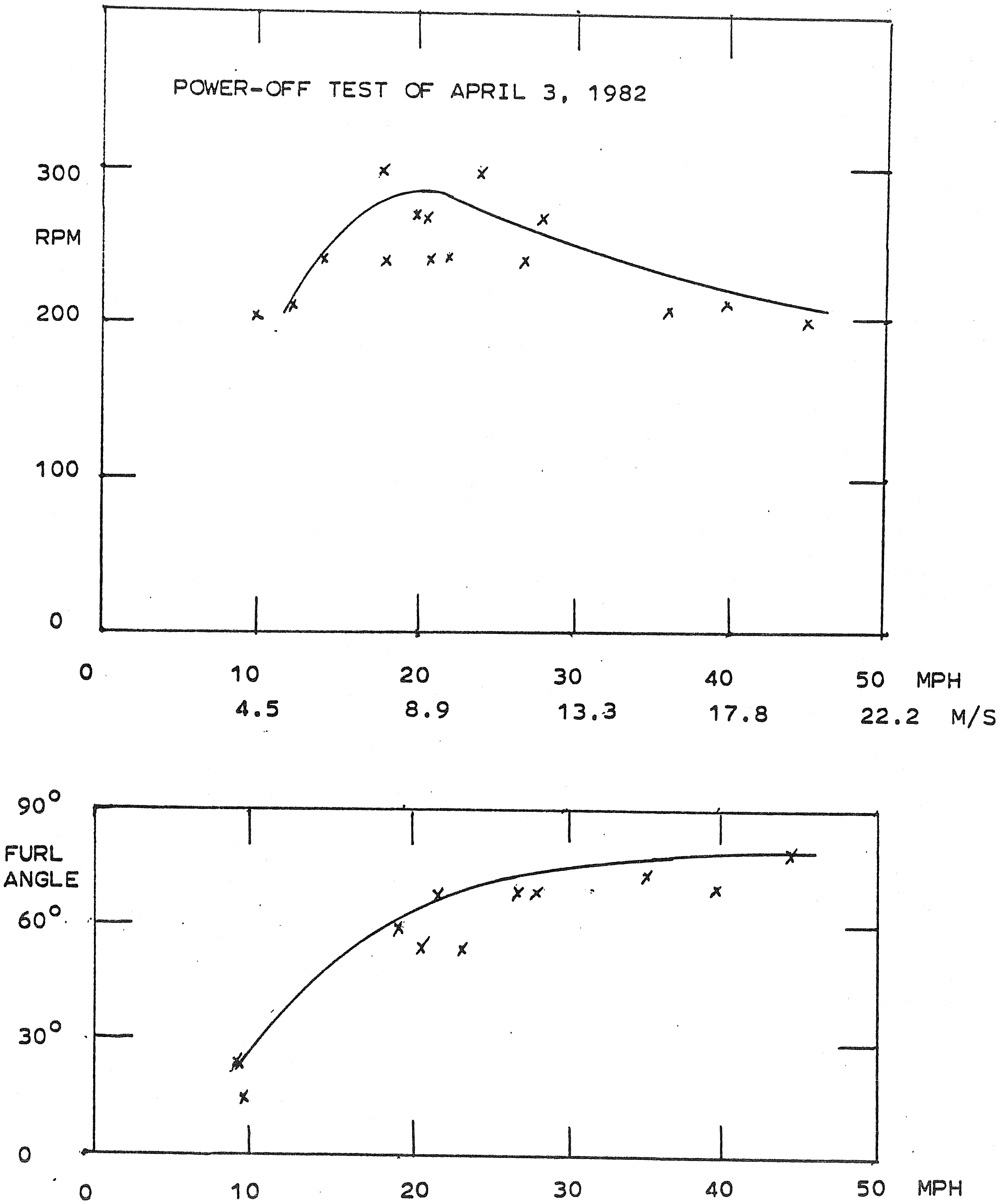


FIGURE 3-3. POWER-OFF RPM AND FURL ANGLE FROM TRIM ANALYSIS (SOLID LINE) AND FROM TEST DATA

springs ($P = 1.0$) the yaw damping moment was negative and that a certain P-value was needed to obtain zero yaw damping. For $\delta_3 = 0$ this value was $P = 1.035$. For $\delta_3 = 67$ deg this value was $P = 1.09$. The actual spring moment was the same in both cases as can be seen from the lower graph of Fig. 3-4. $K\beta$ is the amplitude of the teeter spring moment, $2\ I\Omega\dot{\chi}$ is a reference moment used to non-dimensionalize $K\beta$. The line of zero yaw damping is horizontal which means that the teeter spring moment for zero yaw damping was independent of δ_3.

At operational rotor speed the configuration with the tension teeter springs at Tyson ($\delta_3 = 67$ deg) had a value of $P = 1.022$ and with tension plus compression teeter springs a value of $P = 1.051$. Thus, according to the analysis, the rotor yaw damping was still negative with both sets of teeter springs. The results of a simplified analysis of the teeter amplitudes from rotor yaw rates were presented in Ref. 6 and compared with Rocky Flats test results using the Proof-of-Concept PCP wind turbine which had no teeter springs. For low wind speed, analysis and test results were in reasonable agreement.

Fig. 3-5 taken from Ref. 5 shows for the same parameters listed in Fig. 3-4 the rotor yawing moment vs. steady yaw angle for two P-values (1.0 and 1.05) and for two δ_3 values (0 and 67 deg). Negative moments indicate negative position stability when the yawing moment tends to increase the yaw angle. At $\delta_3 = 0$, approximately neutral yaw position stability was obtained for $P = 1.05$, a value which according to Fig. 3-4 would also eliminate negative yaw damping. At $\delta_3 = 67$ deg, a higher P-value was needed. A value of $P = 1.09$ which removed negative rotor yaw damping should come close to removing also negative yaw position stability. At the maximum P-value achieved with the Tyson machine, $P = 1.051$, Fig. 3-5 shows negative rotor yaw position stability.

It should be noted that neither Fig. 3-4 nor Fig. 3-5 included yawing moments from rotor axis lateral off-set and from rotor axis uptilt. For the rotor yaw damping data of Fig. 3-4 this should lead to only small corrections since neither the mean thrust nor the mean torque were much affected by the yaw rate. However, thrust and torque are greatly affected by rotor yaw angle so that including thrust off-set and torque vector up-tilt angle, Fig. 3-5 would change drastically. The trim analysis results shown in Fig. 3-1 included the moments of Fig. 3-5 and indicated that for fixed tail vane yaw angle the nacelle yaw position stability was positive as explained in the preceding subsection.

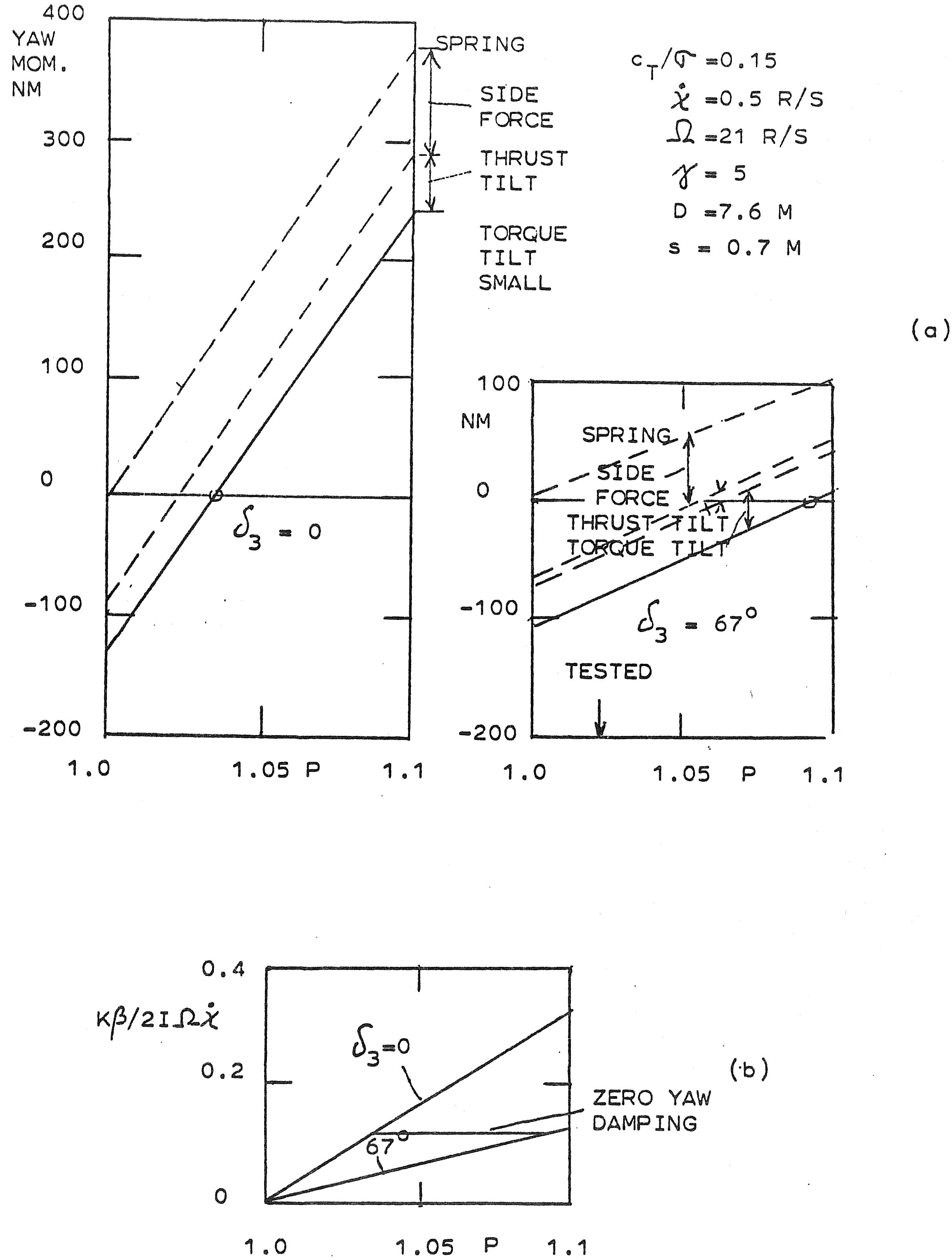


FIGURE 3-4. ROTOR YAW DAMPING MOMENTS (a) AND NON-DIMENSIONAL HUB MOMENTS (b) VS. TEETER SPRING STIFFNESS P - 1

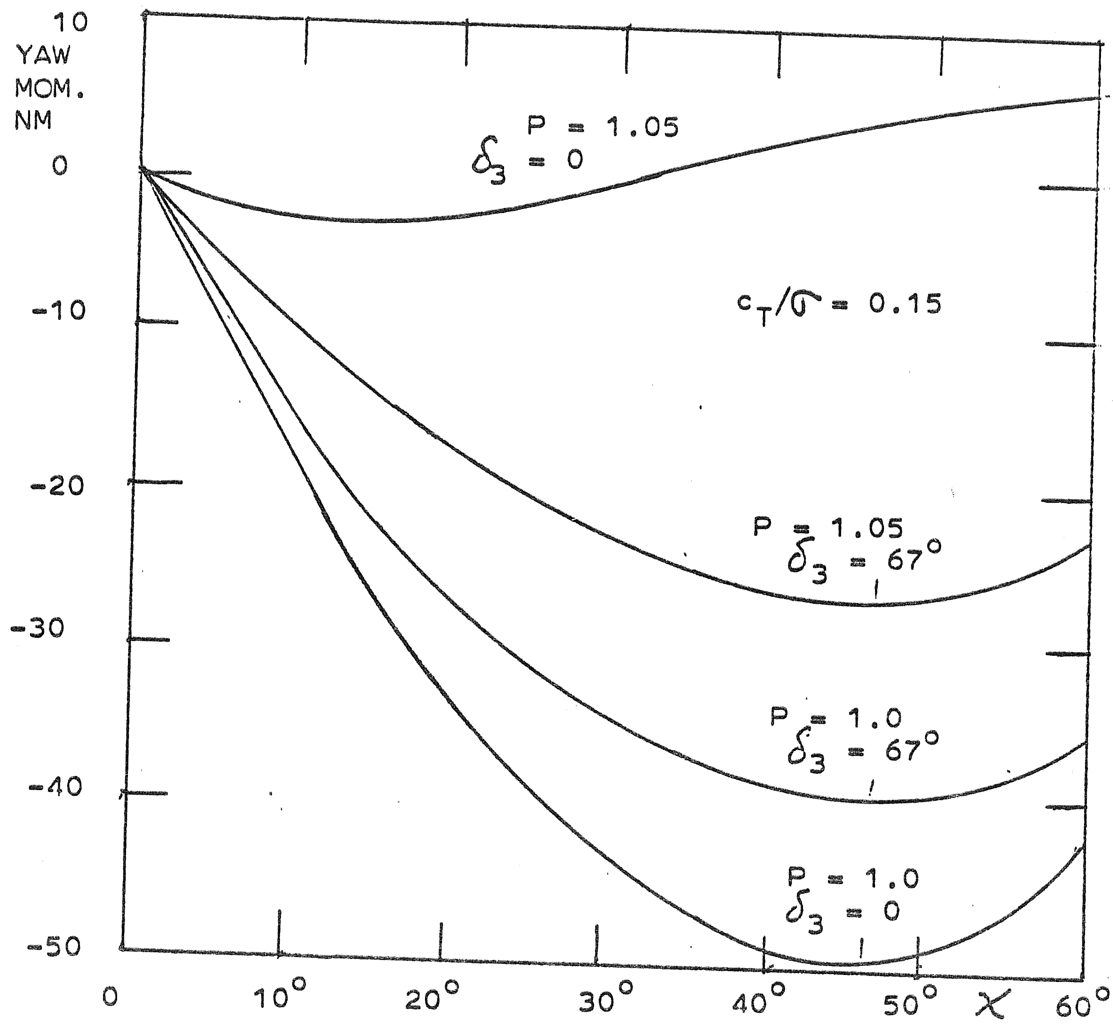


FIGURE 3-5. ROTOR YAWING MOMENT VS. YAW ANGLE

SECTION 4.0

TEST DATA

The modified experimental wind turbine was tested for 24 days between January 1982 and January 1983. The number of tower raisings since May 1980 increased from 72 to 99, an addition of 27 (not all raisings were followed by a test). The number of operational hours since May 1980 increased from 141 to 190, an addition of 49 hours.

The Proof-of-Concept wind turbine was tested for 31 days between April 1983 and October 1985. The number of tower raisings since May 1980 was 127, an addition of 28 since installing the Proof-of-Concept turbine, the number of operating hours was 113 for this machine, of which 60 were unattended.

4.1 EXPERIMENTAL TURBINE

Most of the 24 runs with the modified experimental turbine showed results that were affected by some mechanical deficiencies or by some inappropriate settings of parameters like furl angle range, uptilt angle of the rotor axis, spring preload, damper rate etc. The removal of the large furl bearing friction by the modification to Fig. 2-1 (b) configuration had a very substantial effect and required a stiffer damper and a higher preload of the unfurl spring. The furling motion lost its jerkiness and became smoother, though initially there was excessive break-out friction of the Enidine damper. The early tests of the modified turbine showed severe instabilities from jack knifing of the actuator-damper unit. The phenomenon was removed in several steps by first blocking the pin movement of the clevis between actuator and damper and by finally inserting the rigid coupler shown in Fig. 2-12. The deficiencies caused by the original tail vane and boom (erratic yawing power-on at low wind speed, 1P vertical tail resonance, 4 P bursts of rotor torque and nacelle vibrations at high power) were discussed in Section 2 and were corrected in September 1982 by installing a larger tail vane and a longer tail boom.

4.1.1 Analog Data

Power-off, the passive furl control system worked well and behaved approximately as predicted by the trim analysis. This can be seen in Fig. 3-3 where the power-off analytic trim values for RPM and furl angle are compared to test data taken on April 3, 1982. The wind was extremely turbulent and sometimes fluctuated between 20 and 50 mph (9 and 22 m/s) within two seconds. Nevertheless, the furl angle and RPM variations were small even in severe gusts. The blade loads were also acceptable. From these tests benign storm survival in autorotation seems to be assured.

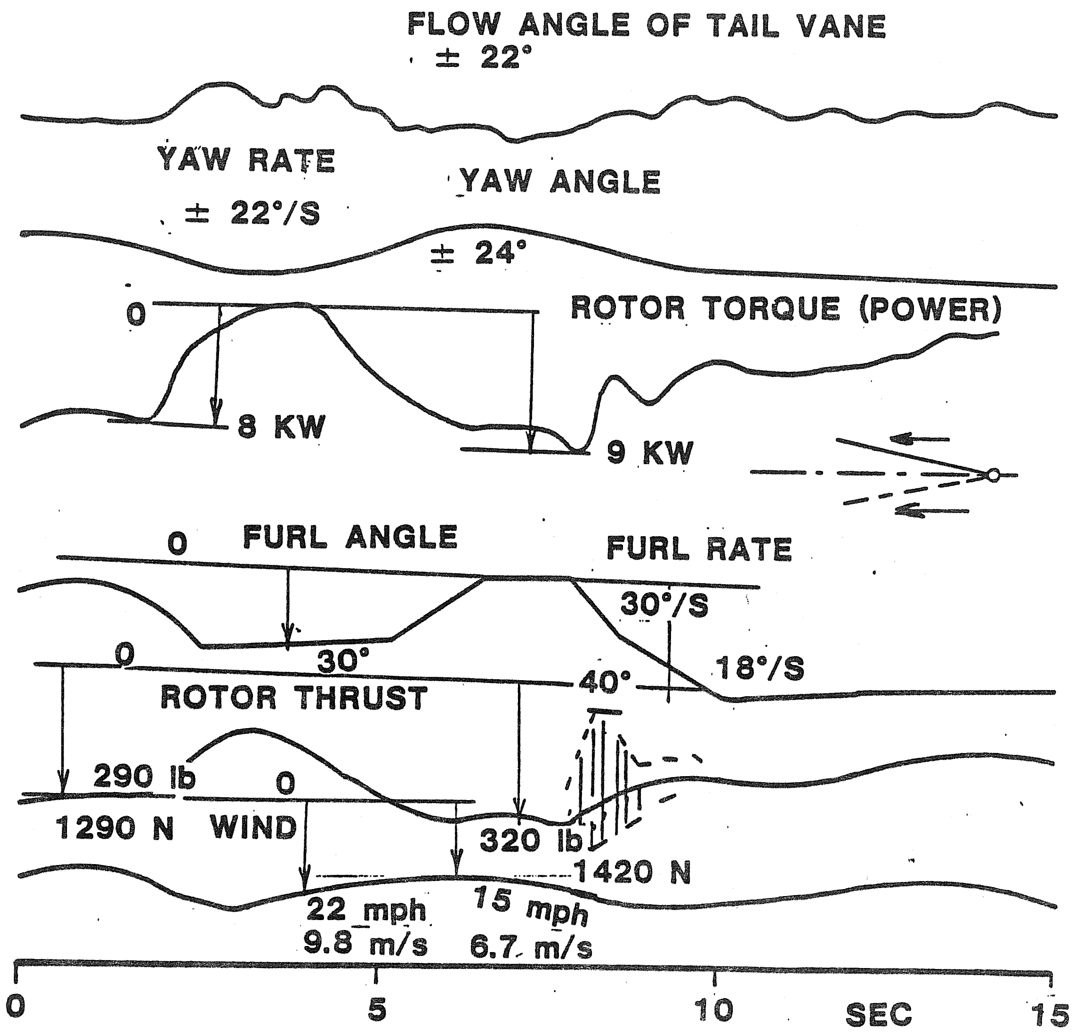


FIGURE 4-1. OSCILLOGRAPH RECORD FOR YAW OSCILLATION, OCTOBER 20, 1982

The power-on behavior in turbulence was, however, less satisfactory. A typical power-on time history taken from Ref. 4 is shown in Fig. 4-1 and was measured on October 20, 1982. The two top curves represent flow angle relative to the vane and vane yaw angle relative to the mast. They are in phase and have about the same amplitude indicating that the wind direction was approximately steady. The lowest curve represents the wind velocity measured 17 ft (5.2 m) below the rotor center. The wind speed oscillated between 15 and 22 mph (6.7 and 9.8 m/s) with a period of about 6 seconds which was the natural period of the tail vane. This explains the large tail vane yaw amplitude without wind direction change. The furl angle time history is somewhat uneven because the damper had at that time a large break-out friction force. However, it is clearly seen that on-set of furling occurred at the time of maximum yaw vane furl rate, and on-set of unfurling occurred at the time of maximum vane yaw rate in the unfurl direction. The rotor yaw angle change from combined tail vane yawing and rotor furling was $\pm$ 50 deg and explains the large fluctuations in rotor torque and rotor thrust. As mentioned before, according to the analysis, rotor torque and thrust are mainly a function of rotor yaw angle, not of rotor yaw rate. The rotor power varied between 0 and 9 KW, the rotor thrust varied between 100 and 300 lb (450 and 1340 N). In contrast, power-off tests showed that tail vane yawing and rotor furling were mostly uncoupled. Apparently, the variable rotor speed and the lower negative rotor yaw damping reduced the coupling between vane yawing and rotor furling.

Many parameter changes were tried to obtain a better power-on turbulence response. Neither changes in uptilt angle of the rotor axis in the range of 7 to 10.5 deg nor changes in the unfurl spring preload and in spring kinematics had an effect on the basic dynamic behavior, though the proper combination of spring preload and uptilt angle was important for the energy capture. The effects of teeter springs and of damper rates will be discussed later.

4.1.2 <u>Wind Speed at Tail Vane</u>

The test results shown in Fig. 4-1 were obtained with the larger tail vane and the longer tail boom. Except for the effects mentioned before, furl dynamics was not affected by this modification. Wind speed measurements with a second anemometer on top of the tail vane installed in May 1982 have shown that unfurled power-on operation reduced the mean wind speed at the tail vane by a factor of 0.7. With non-rotating furled rotor the tail vane anemometer indicated at 11 mph (5 m/s) wind speed the same mean value as the 22 ft (6.7 M) lower tower anemometer. For higher speed, the upper anemometer indicated a lower mean speed than the tower anemometer which amounted to a reverse wind speed gradient as compared to that in the earth's boundary layer. The reason was the potential flow perpendicular to the ridge on which the turbine was located. All tests were conducted in westerly winds which would produce this ridge effect. Due to this effect, the measured wind speeds in the test regime were higher than at the rotor center so that the performance data were conservative. The tail vane anemometer was not normally used and all data were correlated with the lower anemometer.

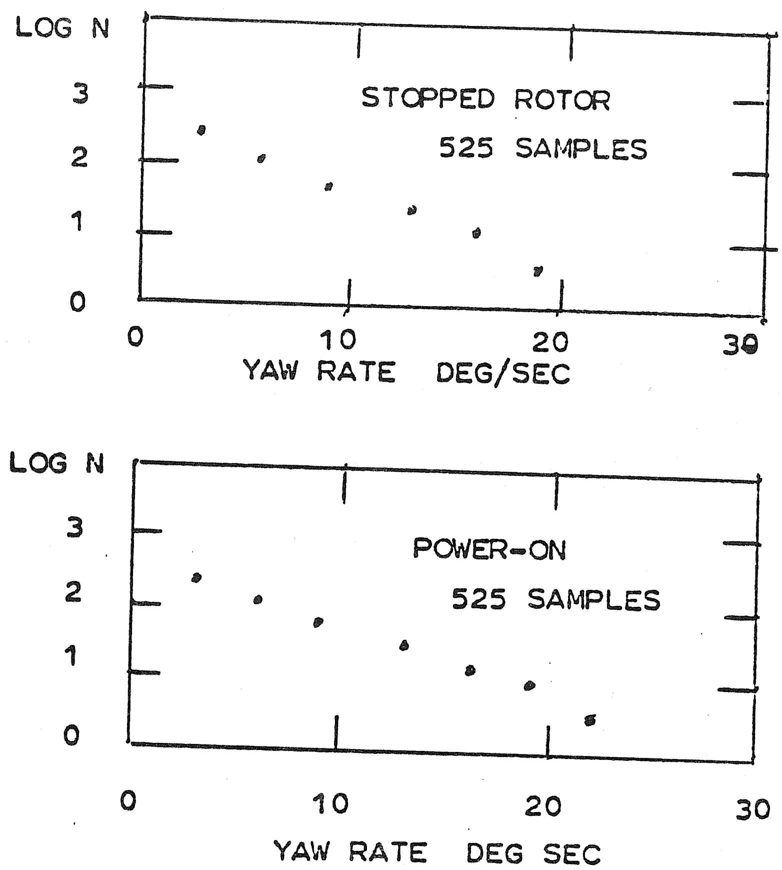


FIGURE 4-2. YAW RATE DISTRIBUTION, NOVEMBER 12,1982

4.1.3 Statistical Data

While in the case represented by Fig. 4-1 the periodic wind speed variations were the cause of the tail vane oscillations, statistically this appears to be a rare event as can be concluded from Fig. 4-2 also taken from Ref. 4. On November 12, 1982, the yaw rate distribution of the tail vane was measured with stopped rotor and in power-on operation. The two distributions were almost the same. Thus the off-set rotor thrust seemed to have little effect on the yaw oscillations which were statistically mainly caused by wind direction changes.

A typical statistical evaluation of a test run made on October 10, 1982 is shown in Fig. 4-3, also taken from Ref. 4. The left upper graph shows the wind speed distribution in terms of log N (N is the number of samples per wind speed bin). The other three graphs give means and standard deviations of furl angle, rotor power (computed from the measured rotor torque and rotor speed) and cyclic pitch amplitude for each wind speed bin. While this run was made in a condition of moderate turbulence, Fig. 4-4 shows the measured furl angle means and standard deviations for a high turbulence test on November 12, 1982. Both mean and standard deviation for the furl angle are substantially higher for the same wind speed bins.

The data shown in Figs. 4-3 and 4-4 made use of a machine language program for the Apple II computer. 128 samples were taken during one second about every three seconds and assigned to a wind speed bin from a single wind speed sample taken at the beginning of the sampling period. Mean and standard deviation for each wind speed bin were updated for each sample set. The problem with this method was that the measured wind speed at the anemometer site was not the average wind speed that the rotor experienced. In Ref. 7 it is shown that preaveraging of the data before application of the method of bins will low-pass filter the data and reduce the scatter in the bins. The correct preaveraging time is a function of the anemometer location and increases with axial, horizontal and vertical displacement from the rotor center. Also, due to the difference between a point wind speed measurement and the average wind speed over the rotor disk, a filtering effect will occur that can be simulated by a certain preaveraging time. If this time exceeds the correct time, as it usually does in wind systems performance measurements, meaningful data are discarded. If the preaveraging time is less than the correct time, the data contain turbulence inputs to the anemometer that are unrelated to the response of the wind energy system which is the case with the Fig. 4-3 and Fig. 4-4 data. Unfortunately, the computer input data had to be discarded after the means and standard deviations had been determined for each wind speed bin so that there was no way to afterwards correct the data for the effects discussed. However, oscillograph data were preserved and could be preaveraged later. The results of preaveraging will be discussed in Subsection 4.2.3.

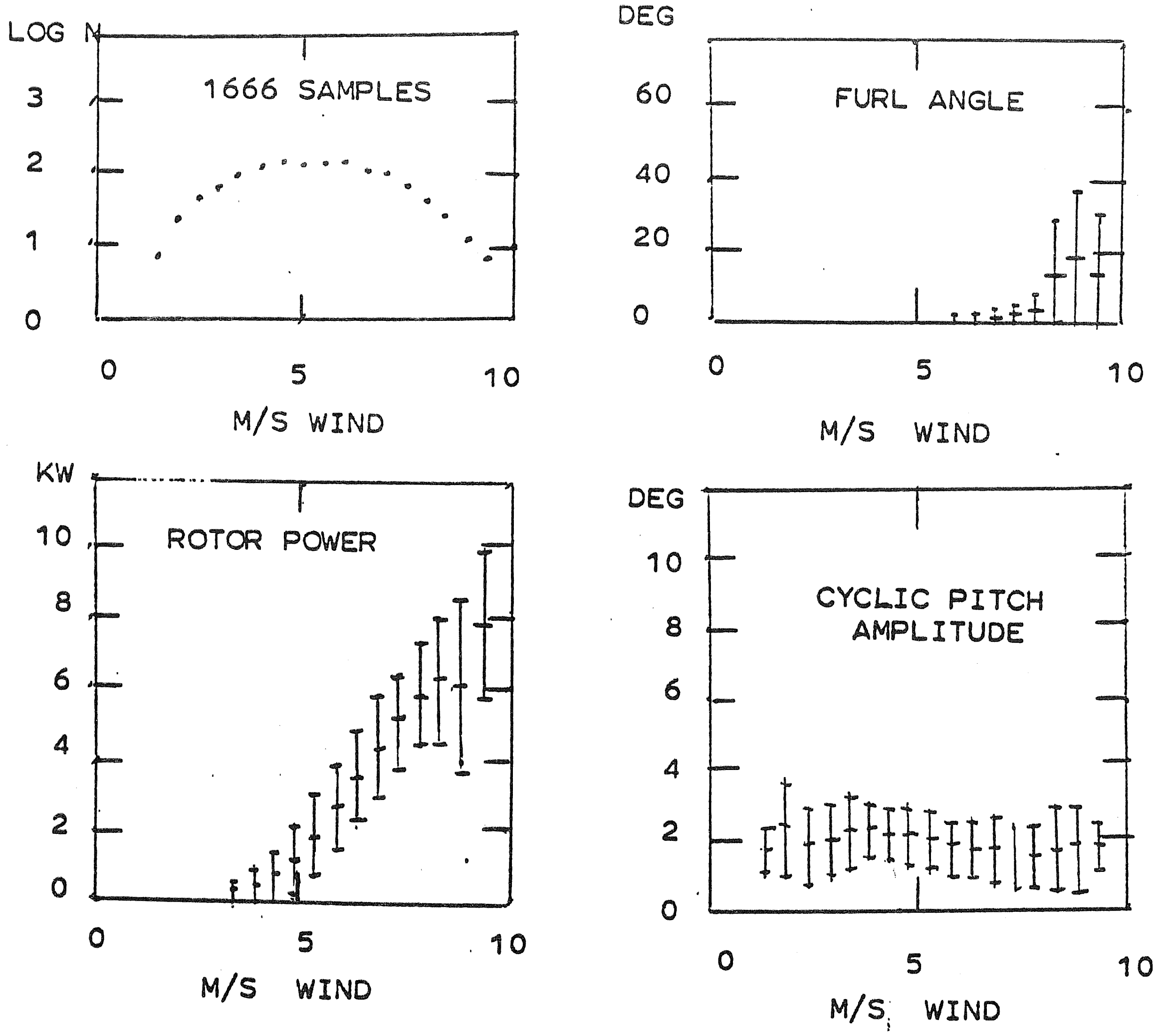


FIGURE 4-3. BIN PLOTS TAKEN OCTOBER 10, 1982

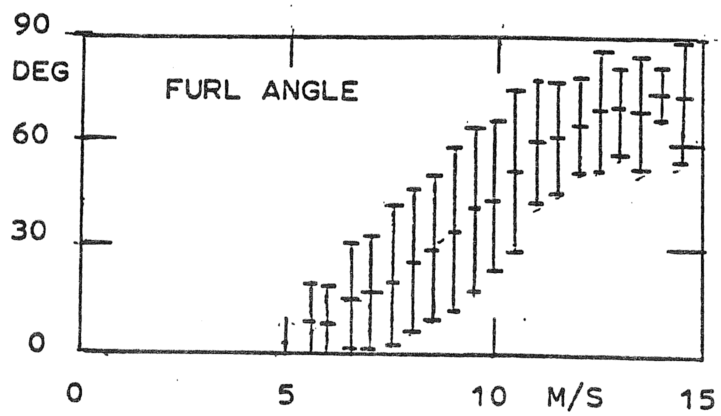


FIGURE 4-4. FURL ANGLE BIN PLOT TAKEN IN SEVERE TURBULENCE NOVEMBER 12, 1982

4.1.4 Tests with Stalled Blades

In December 1982, the transmission ratio was changed from 9.37 : 1 to 12.5 : 1. At the same time the elastic teeter stops were narrowed from $\pm$ 11 deg to $\pm$ 5 deg of free travel to lower the impacts in case of blade stall. The damper was changed to give in the furl direction 50% more damping than in the unfurl direction. This led to a very low maximum furl rate of only 5 deg/s vs. a maximum unfurl rate of 20 deg/s. On January 15, 1983 the rotor reached the stall regime and strong teeter impact loads occurred. As a consequence, the teeter bearings suffered some wear, and the elastic teeter stops were damaged. Blade stall was delayed far beyond the expected limit. Fig. 4-5 shows the torque coefficient c_Q/σ vs. speed ratio $V/\Omega R$. The test points represent mean values from a large number of analog data selected during periods of nearly steady wind speed. The dashed line was obtained from the prop code using two-dimensional airfoil data. Above $c_Q/\sigma = 0.02$ this analysis was in error by a large margin. The solid line was obtained with the NASA stall model described in Ref. 8. In agreement with the test results this model predicted more than twice the maximum rotor torque.

Fig. 4-6 shows the blade angles of attack obtained with the NASA model for four tipspeed ratios $\Omega R/V$. The tests were conducted up to $\Omega R/V = 3$ where the entire blade is stalled. At $\Omega R/V = 4$ ($c_Q/\sigma = 0.03$) only the inner portion of the blade is stalled. The blade will be nearly unstalled at $c_Q/\sigma = 0.02$ which should be considered a maximum value to be used in teetered rotors. This value corresponds to 7 KW rotor power at 12.5 transmission ratio, and to 17 KW rotor power at 9.4 transmission ratio. Thus the lower gear ratio turbine with a maximum rotor power of 10 KW (see Fig. 4-3) operated unfurled with a wide stall margin.

4.2 PROOF-OF-CONCEPT TURBINE

Many of the 29 test runs with the Proof-of-Concept turbine were also affected by equipment deficiencies and had to be repeated after corrections were made. The first four runs were needed to solve the problems of operating the three phase generator with single phase current, discussed in Subsection 2.2. The emergency shut-down system, particularly the vibration shut-down part, required many adjustment runs before it worked properly. The tower winch failed and had to be replaced. The hydraulic tower lifter failed several times and needed replacements. The electric actuator power unit accumulated water while the tower was down and had to be replaced several times and better sealed. The furl strain gage bridge failed and was replaced. The Vishay strain gage signal conditioner needed repair from time to time. The pinned belt driving the tail vane yaw potentiometer stretched and slipped off the sprockets so that an important piece of data was missing. Several times a suitable wind was not utilized because the Corps test engineer was not available due to higher priority assignments. The first unattended run, October 19 to 22, 1984, was terminated because the anemometer and the furl indicator

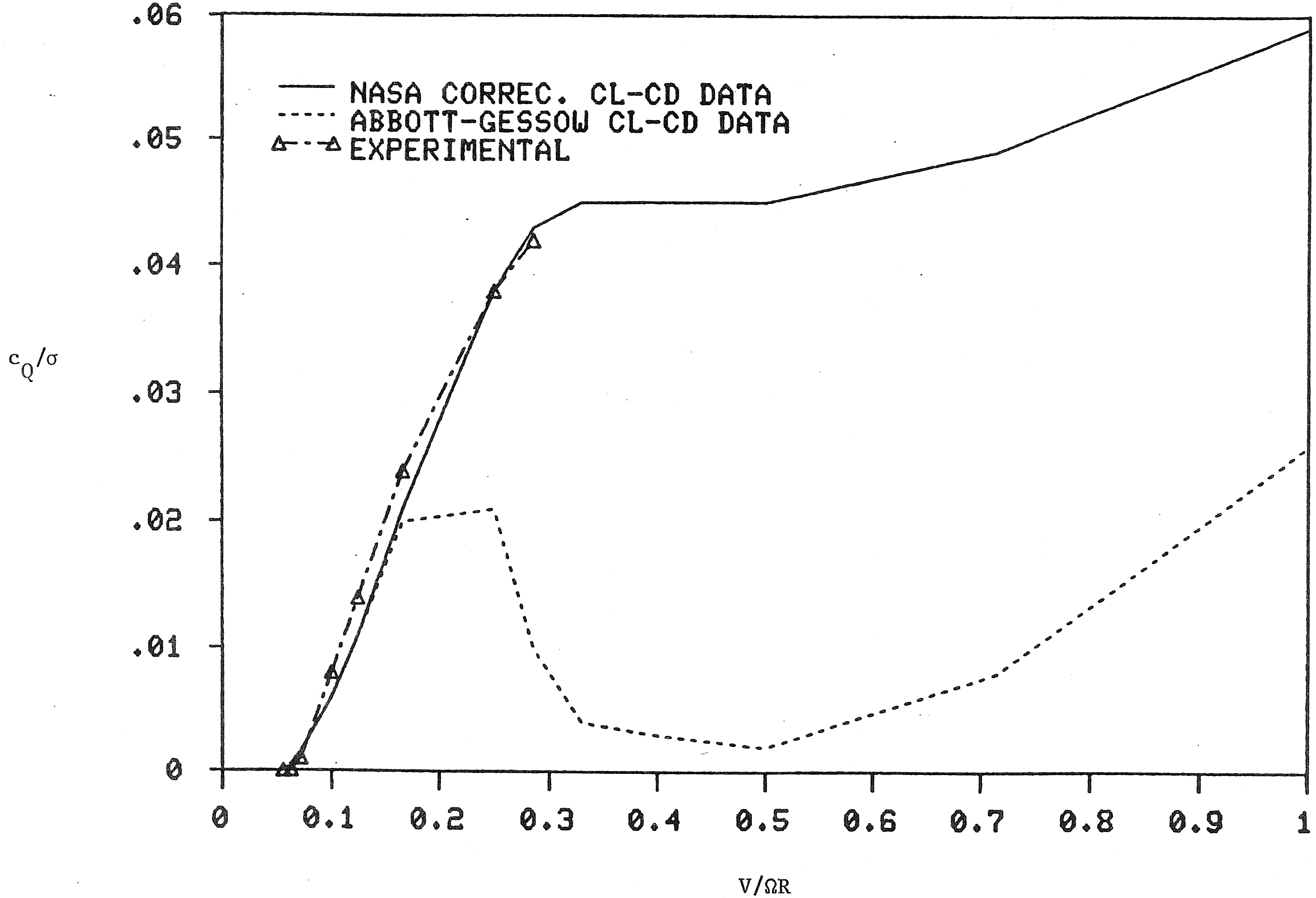


FIGURE 4-5. TORQUE COEFFICIENT FOR UNFURLED OPERATION VS. SPEED RATIO, JANUARY 15, 1983, PROP CODE DOTTED LINE, NASA MODEL SOLID LINE

failed. The second unattended run, October 24 to 27, 1984 was terminated by the tripping of the circuit breaker from generator overload. Due to this overload condition it was decided not to continue the unattended runs. However, the tower was left in the raised position with furled rotor between November 17, 1984 and January 19, 1985. During these two months there was harsh winter weather with high storm velocities. Except for some instrument deterioration no damage was observed. The wire bundle inside the tower was not noticeably twisted.

4.2.1 Vibration Data

Vibration tests with raised tower and non-rotating turbine were conducted in October, 1983. Upon impact excitation of the blades, two natural frequencies were observed in the oscillograph trace of the bed plate accelerometer; 8.0 and 2.0 Hz. The first was the blade coning frequency and it agreed with measurements made earlier with the rotor alone before installation on the tower. The second frequency was that of the fundamental tower bending mode which could also be observed as a 1 P resonance at 120 RPM during rotor start-up. It was not clear why a much lower blade coning frequency of 6.3 Hz had been determined at Rocky Flats.

It was found that the damper configuration had a large effect on nacelle vibrations. The damper force was proportional to the square of the damper rate so that the damper provided at high frequencies a rather rigid connection between nacelle and tail vane boom. The nacelle yawing oscillation, due to the uptilt angles of the rotor and generator axes, produced power-on rotor torque and generator current oscillations. Thus the nacelle yawing vibration was seen not only in the bed plate accelerometer trace but also in the generator current trace. The nacelle natural frequency could be best determined from the motion following an unfurl stop impact. The relation between the damper rate under 50 lb (222 N) force and the nacelle natural yawing frequency is presented in Table 4-1. The dates refer to the test runs with the four different damper inserts.

The frequency of rotor rotation at synchronous generator speed was 3.2 Hz so that the damper with 0.7 inch/s rate caused a 1 P nacelle resonance which resulted in occasional large 1 P current oscillations up to ± 6 amps. Efforts to detune the system by inserting rubber washers into the damper clevis mount were unsuccessful. The stiffer damper configuration with a rate of 0.5 inch/s was used only in April 1984 and resulted in a very low unfurl response leading to a substantial loss of captured energy. The lowest damper stiffness (1.3 inch/s rate) resulted in high furl rates and occasional large amplitude furl oscillations with a period of about six seconds which was the natural period of the tail vane in yaw. Thus the only practical damper settings were those with 1.0 and 0.7 inch/s rates under 50 lb force. Of these two settings only the 1.0 inch/s rate provided a natural nacelle frequency that was not in resonance with 1 P excitation.

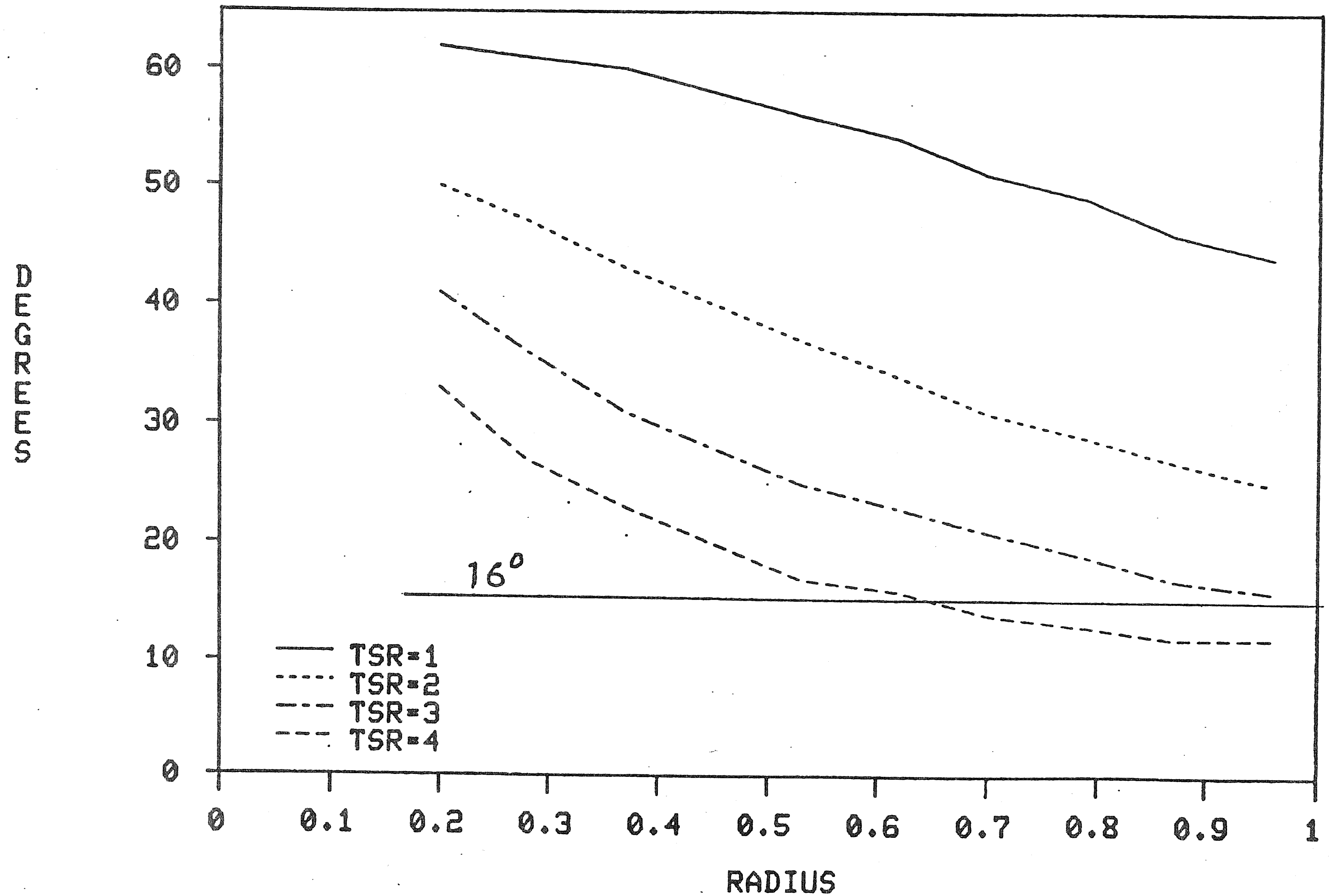


FIGURE 4-6. BLADE ANGLE OF ATTACK VS. NON-DIMENSIONAL RADIUS, NASA MODEL

TABLE 4-1. Damper Rate vs. Nacelle Yawing Frequency

Damper Rate inch/s(50 lb)	Nacelle Yawing Frequency Hz	Date
1.3	1.6	5-19-83
1.0	2.5	5- 2-83
0.7	3.2	5-28-84
0.5	3.8	4-30-84

4.2.2 Effects of Parameter Changes

The evaluation of the various parameter changes for the Proof-of-Concept turbine was made on the basis of the furl characteristics in turbulence. Actually the rotor yaw angle rather than the rotor furl angle should have been used since thrust and torque at a given wind speed are a function of rotor yaw angle. One would need then the measurement of the wind direction with respect to the rotor. Available was only the wind direction at the top of the tail vane which, due to the rotor wake effect, was not the same as that at the rotor. However, in the mean, the tail vane boom assumed approximately the direction of the wind so that the mean rotor furl angle should be close to the mean rotor yaw angle.

For the furl response to turbulence one would like to have neither over-sensitivity leading to large furl and power fluctuations, nor under-sensitivity leading to a delay in response and a loss of energy capture. During the parametric testing both of these conditions were encountered. Increased teeter spring stiffness removed, according to the analysis, some of the negative rotor yaw damping and slowed the furl and unfurl responses. Reduced damper rates also slowed the furl response. Thus one could trade teeter spring stiffness against damper rates and vice versa. Table 4-2 gives some qualitative insight into the teeter spring and damper rate effects. The cited numbers are typical values of furl oscillation amplitudes and maximum furl rates in the wind region between 15 and 25 mph (7 and 11 m/s). The last column shows whether the response was judged to be either over or under-sensitive. The numbers in table 4-2 should be taken only as trend indicators, they have no absolute significance. It is seen from Table 4-2 that for each stiffening of the teeter springs at a given damper rate the trend is the same as for a reduction in damper rate at a given tester spring stiffness.

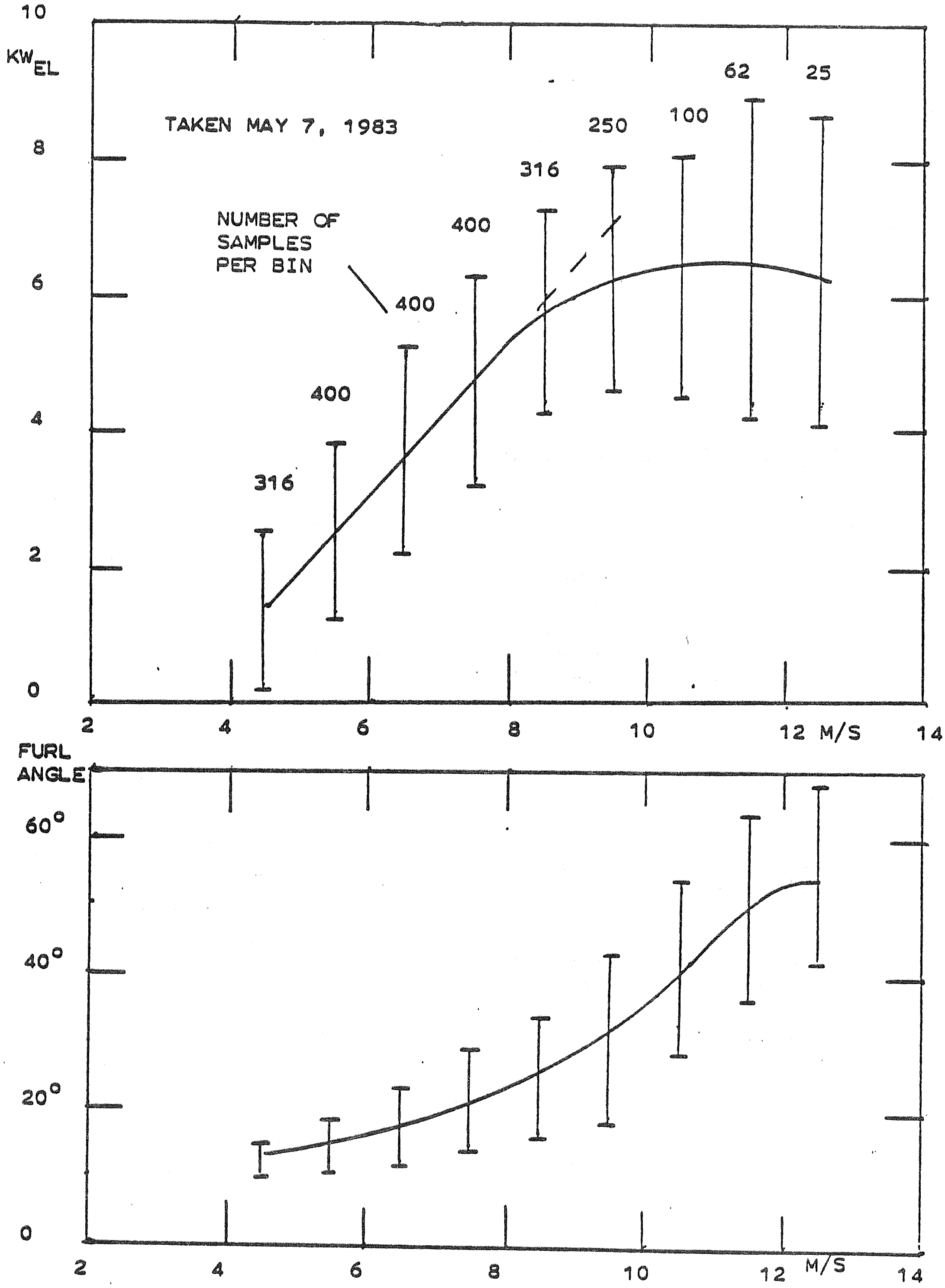


FIGURE 4-7. BIN PLOTS FOR ELECTRIC POWER OUTPUT AND FURL ANGLE, MAY 7, 1983

TABLE 4-2. The Effect of Teeter Spring Stiffness on Damper Rate and Furling Characteristics

Date	Teeter Spring Ft lb/Deg	Damper Rate inch/s	Furl Amplitude	Max Furl Rate °/s	Over Sensitive	Under Sensitive
5-2-83	20	1.0	± 22°	23	*	
5-7-83	27	1.0	± 13°	20		
5-19-83	27	1.3	± 22°	25	*	
5-22-83	45	1.3	± 17°	22	*	
4-6-84	75	0.5	± 3°	6		*
4-30-84	27	0.5	± 6°	10		*
5-28-84	27	0.7	± 12°	18		
10-4-85	62	0.7	± 6°	10		*
?	62	1.0	?	?		

The preload of the unfurl spring had to be changed a number of times. Due to different combinations of teeter spring stiffness and damper rate, the furling system became more or less sensitive to gusts. The unfurl spring load had to be adapted to the changing sensitivity to avoid generator overload or loss of energy capture. A good criterion for the proper unfurl spring adjustment was the maximum autorotational RPM as seen in Fig. 3-3. This maximum should be at 250 to 270 RPM. In this range reasonable power-on characteristics were obtained.

4.2.3 Statistical Performance Data

Fig. 4-7 shows bin plots of furl angle and of electric power output for the test of May 7, 1983. The high turbulence level during the test caused high mean furl angles and a low maximum mean power output of 6.4 KW_{EL}. Fig. 4-8 shows the result of a test conducted May 28, 1984. A stiffer unfurl spring had been installed and adjusted such that the spring unfurl moment over the entire furl angle range was larger by a factor of 1.25 as compared to the May 7, 1983 test shown in Fig. 4-7. The damper insert with the flow passages had been changed to obtain 1.4 times the damper force at a given rate (0.7 inch/s or 0.018 m/s under 50 lb or 222 N force). The mean furl angle was for a given wind speed bin lower than before, the electric power output somewhat higher, particularly in the upper wind speed range. How much of this change

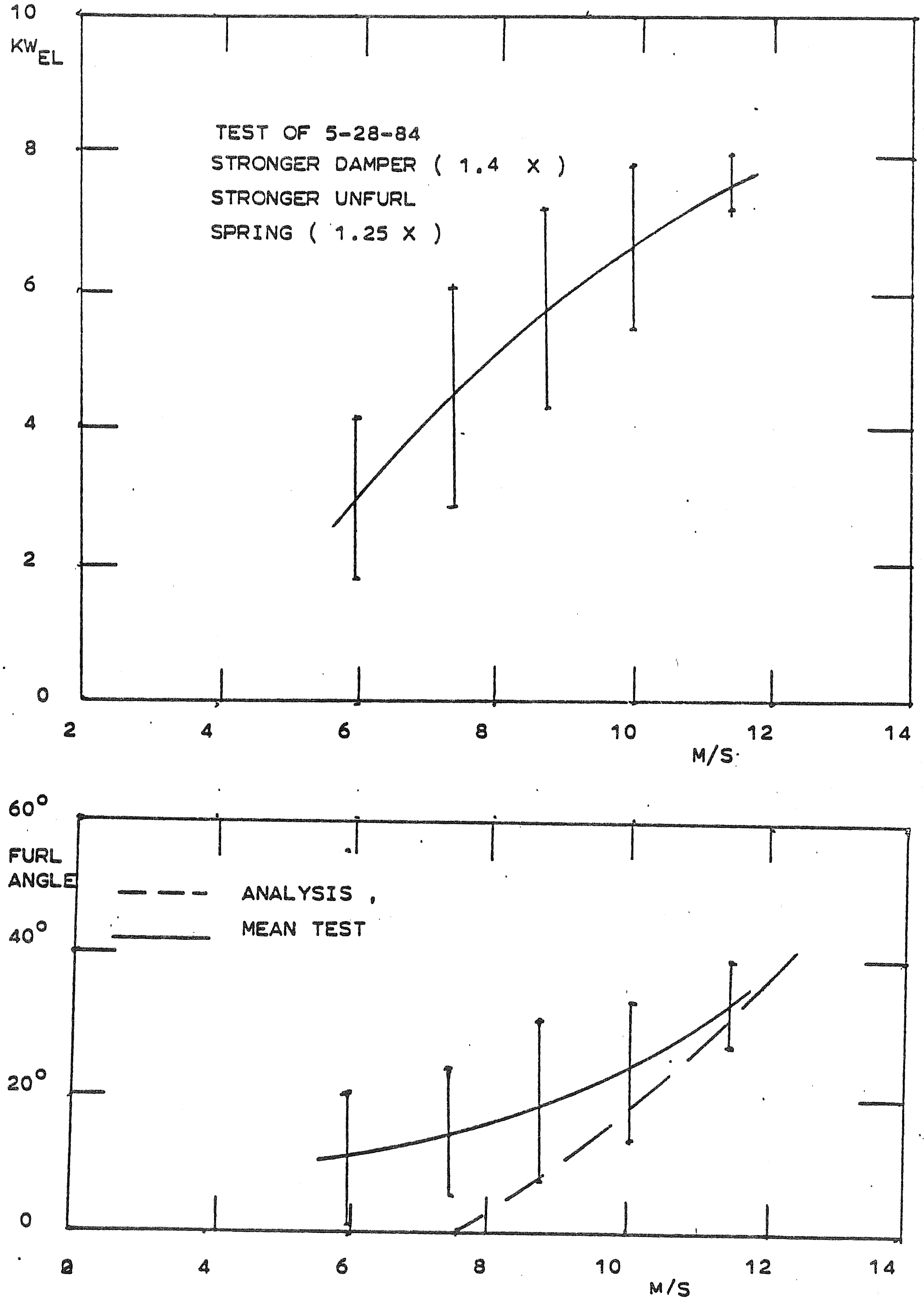
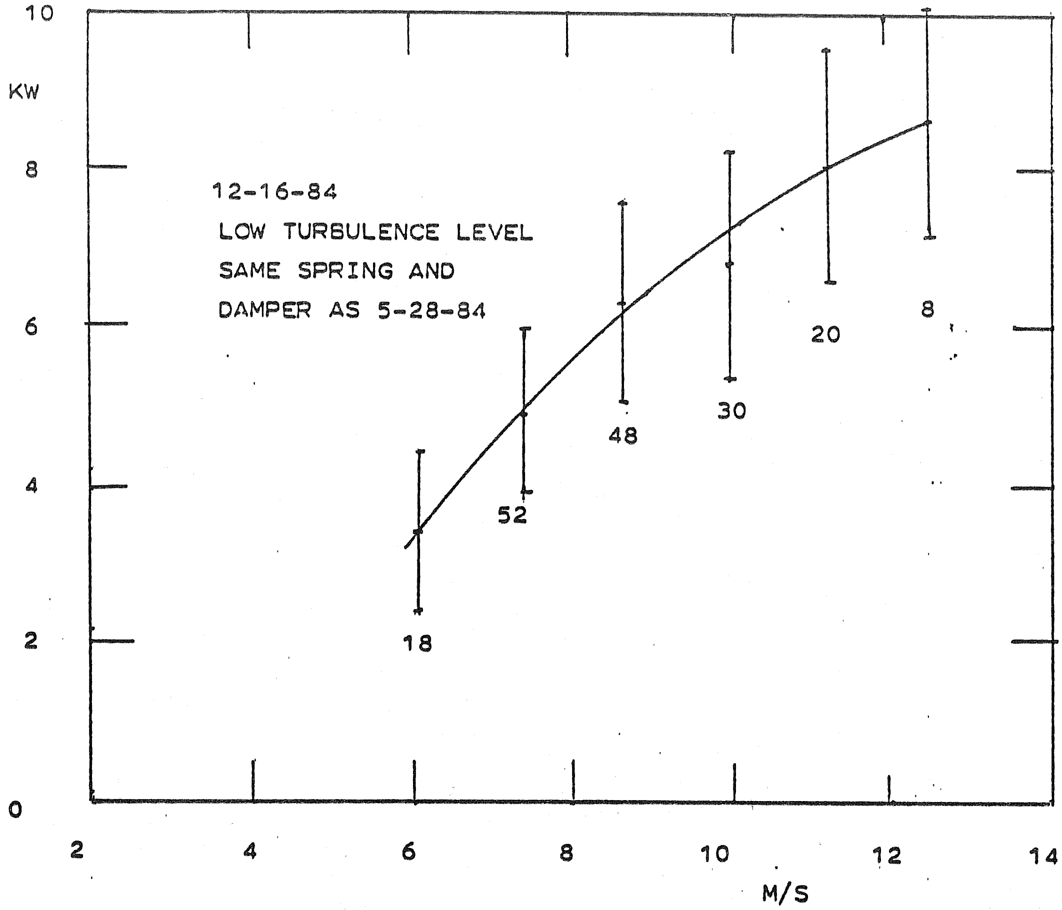
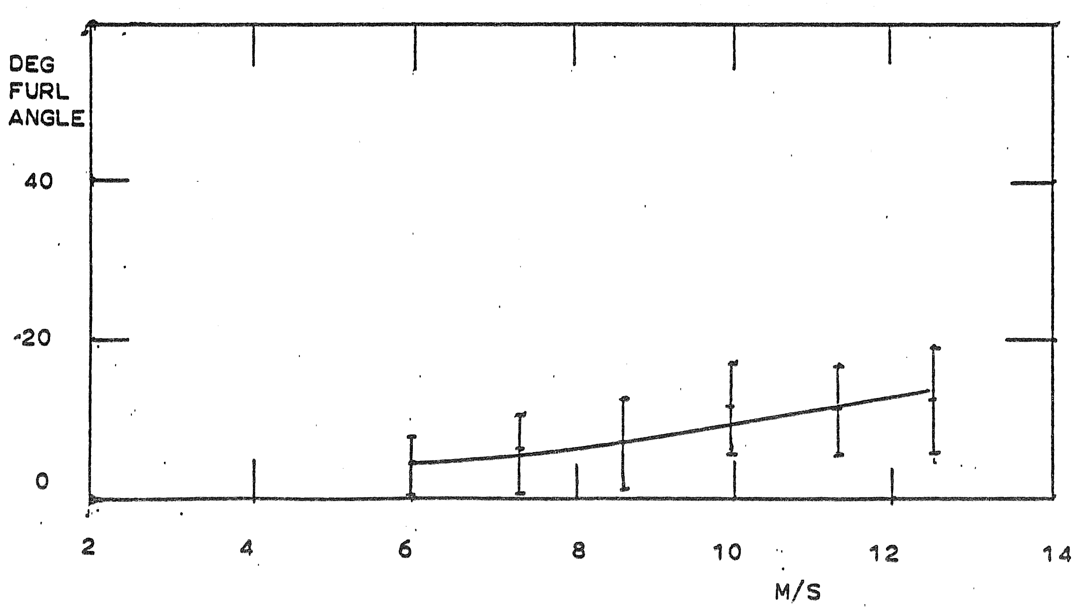


FIGURE 4-8. BIN PLOTS FOR ELCTRIC POWER OUTPUT AND FURL ANGLE, MAY 28, 1984

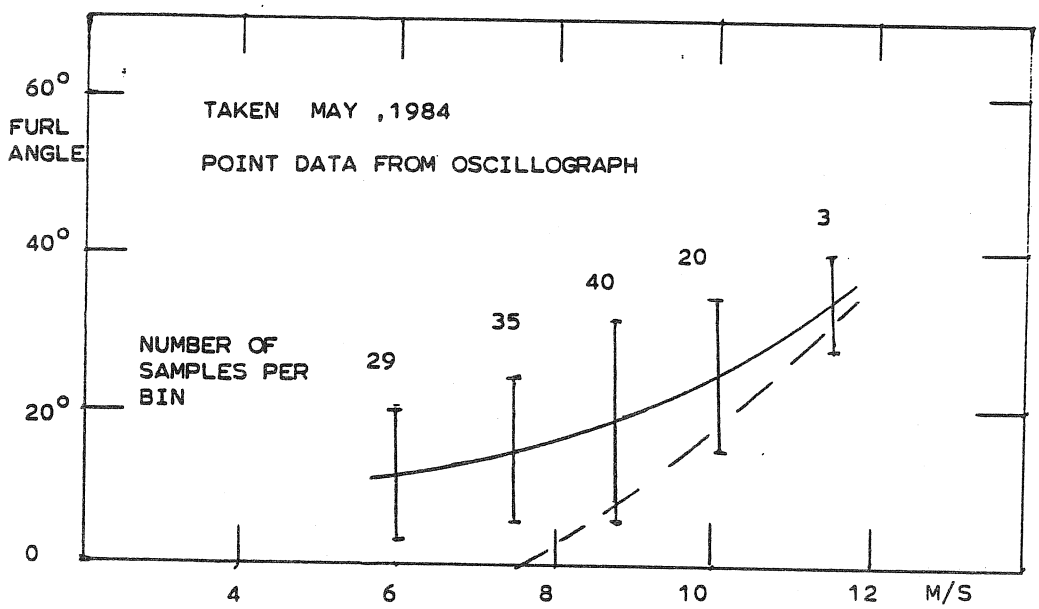
was due to the stronger spring and damper and how much to a possibly lower turbulence level was difficult to judge. The dashed line in Fig. 4-8 represents the analytical prediction of furl angle vs. wind speed taken from Fig. 3-1. The measured mean furl angle is higher than the predicted furl angle particularly for lower wind speed. Fig. 4-9 obtained on December 16, 1984 shows a low turbulence case where the mean furl angle was lower and the electric power output was higher than for the test of 5-28-84 shown in Fig. 4-8. The mean electric power reached 9 KW and the upper standard deviation 10 KW.

Following Ref. 7, the data from the tests of May 28, 1984 were reevaluated with a four second preaveraging time. Fig. 4-10 shows the effect of preaveraging on the furl angle. The upper graph, identical to that of Fig. 4-8, was the result of point samples, the lower graph was the result of preaveraged samples. Though the number of samples in each wind speed bin was rather small in both cases, the effect of preaveraging was clear. Not only the standard deviations were reduced but also the mean values. The associated mean electric power output for a given wind speed was somewhat increased when using preaveraging. The result obtained by preaveraging with a four second time interval should be more realistic than the result from point sampling, which was polluted by the high wind speed frequencies experienced by the anemometer but not by the rotor. Despite this improvement of the statistical data, the basic response to turbulence shown in Fig. 4-1 cannot be smoothed by any method of data processing. It appears that the root cause of the furl and power fluctuations in turbulence is the fast responding tail vane oscillating in phase with the furl angle so that the rotor yaw rate was made up of the sum of tail vane yaw and rotor furl amplitudes. The obvious solution is to slow down the motion of the wind following element; for example, by replacing the tail vane with a slow acting yaw control side rotor. For an active yaw control system with a yaw gear drive the problem would not occur.





FIHURE 4-9. BIN PLOTS FOR ELECTRIC POWER OUTPUT AND FURL ANGLE, DECEMBER 16, 1984



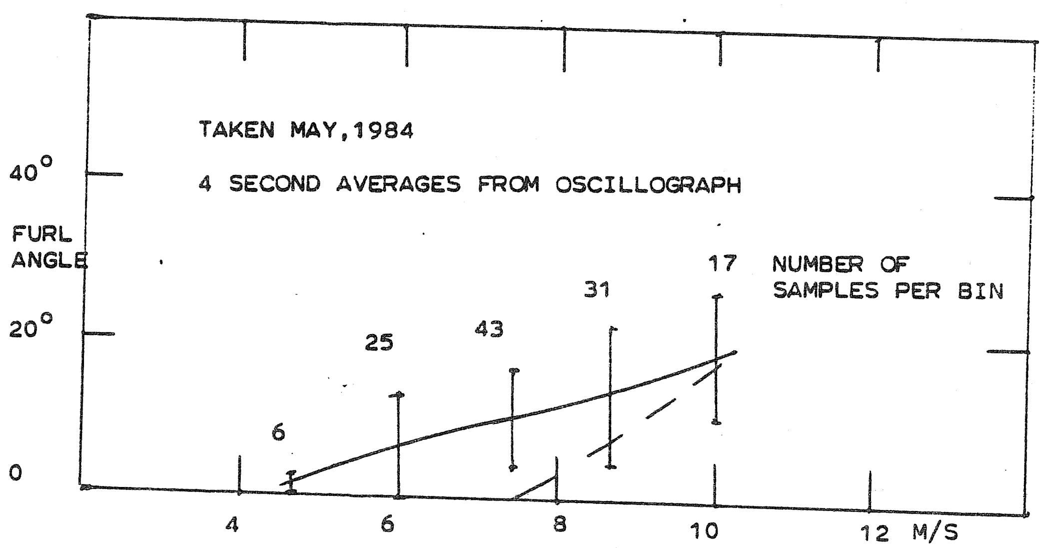


FIGURE 4-10. FURL ANGLE BIN PLOTS WITHOUT AND WITH DATA PREAVERAGING

SECTION 5.0

CONCLUSIONS

Analysis and atmospheric experiments with a 25 ft (7.6 m) diameter tail vane stabilized PCP wind turbine having passive rotor furl control, conducted during the period of January 1982 to October 1985, allowed the following conclusions.

- The modification to independent yaw and furl controls was successful in eliminating delayed furling and unfurling from high bearing friction.
- The blade modification to high twist high chord root sections was successful in causing early start-up.
- The 50% increase in tail vane area and 12% increase in tail boom length was successful in eliminating erratic power-on low wind speed yawing and in correcting two dynamic conditions.
- Increasing the transmission ratio from 9.4 to 12.5 allowed operation in deep stall with strong teeter stop impacts. The torque was twice that predicted by the conventional prop code.
- The modification to the Proof-of-Concept turbine where the shaft mounted gear box and the belt driven generator were replaced by an integrated gear-generator unit, further reduced the start-up wind speed because of an 80% reduction of the break-out torque.
- The trim analysis predicted in agreement with the tests that the power-off rotor speed would peak at 20 mph (9 m/s) wind velocity and that it would drop off at lower and at higher wind speed. From tests with up to 60 mph (27 m/s) gusts, benign storm survival in autorotation can be expected.
- The trim analysis predicted power-on lower furl angles than the measured means.
- Turbulence increased the standard deviations and the means of the furl angle. It decreased the mean electric power output.
- The rotor yaw damping analysis predicted in qualitative agreement with the tests a reduction in negative rotor yaw damping with increasing teeter spring stiffness.
- Damper characteristics were found to strongly influence furl control dynamics. Damper rates of 0.7 to 1.0 inch/s (0.018 to 0.025 m/s) under 50 lb (222 N) force were optimal for teeter spring stiffnesses of 30 to 60 ft-lb/deg (40 to 80 Nm/deg).
- Optimal unfurl spring settings varied with furl friction, damper setting and teeter spring stiffness. Typical values were 150 lb (667 N) unfurled to 220 lb (980 N) furled.
- Preaveraging the test data before performing a bin analysis reduced the furl and power standard deviations, decreased the mean furl angle and increased the mean power output in a given wind speed bin.

To answer in a more generic way the question whether rotor yaw control can be advantageously substituted for blade pitch control, the following points should be considered.

- A rotor yaw control dynamics analysis is needed for a teetered rotor including the effects of partial blade stall.
- The present PCP wind turbine is not well suited for verifying a rotor yaw control dynamics model because of the complex and unknown aerodynamic rotor-tail vane interaction.
- Slowing the response of the wind following element, for example by replacing the tail vane with a yaw control side rotor, should make the turbine suitable for verifying a rotor yaw control dynamics model which could then be used for a generic study of rotor torque and speed control by yawing.

SECTION 6.0

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15. Supplementary Notes

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16. Abstract (Limit: 200 words)

This report describes research conducted at Washington University to address whether or not conventional blade-pitch control of wind turbines can be replaced by rotor yaw control. The test equipment consisted of a horizontal-axis wind turbine, mounted on a 60-ft tower, with an upwind, vane-controlled, passive-cyclic-pitch (PCP) rotor 25 ft in diameter. The automatic rotor-control system was passive. Some erratic yawing at low-speed, power-on operation was corrected with a larger-area tail vane and a longer tail boom. The turbine was tested at gusts of up to 60 mph. The power-off rotor speed was always kept fairly constant. The results would indicate smooth low-load storm survival. Generator cut-in and cut-out were achieved with a digital controller responding to rotor speed. Torque was found to peak at twice the value predicted by the conventional prop code. Overall, the PCP rotor continued to function very well, even at high yaw rates and wind velocities. However, the combination of the passive furl control system and a fast-reacting, wind-following tail vane caused some problems that could not be corrected at the time.

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